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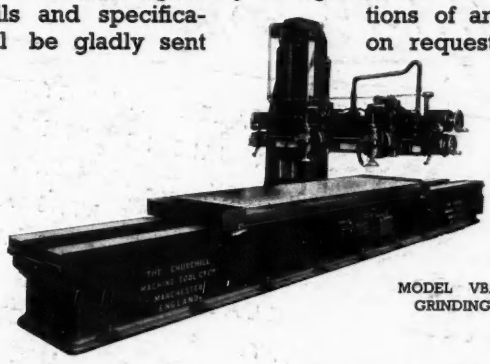

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Industrial Physiology

By D. B. DILL. (From *The Engineers' Digest*, American Edition, Vol. 5, No. 2, February, 1948, pp. 41 and 91).

PHYSIOLOGY is one of our oldest medical sciences; while sometimes described as the handmaiden of medicine, it might better be called the mother of medicine. Certainly it has an intimate relation to all the medical sciences, having, for example, such relatively youthful offspring as biochemistry, biophysics, pharmacology, and psychology.

Industrial physiology is an area of physiology that is concerned with man at work—the processes whereby he converts the chemical sources of energy in his food to useful mechanical work and into a much larger proportion of heat. Incident to this basic function, body temperature must be thermostatically regulated and excess heat dissipated. This much was known one hundred years ago as a result of the classic studies of Lavoisier late in the eighteenth century and of the subsequent research that led to the formulation of the law of conservation of energy. The extremely complicated mechanisms that make these processes effective have been partially revealed by physiological research since then. It is the purpose of this review to point out some of these recent developments in physiology that concern management and the worker.

Energy Exchange. Converting the energy of food to mechanical energy is at best 20 per cent efficient. The by-product heat serves to maintain body temperature, but in a high environmental temperature its dissipation is a burden. The total energy requirement covers the basal metabolic rate or BMR (about 1,800 calories* per day or 75 per hour) which is required to keep the bodily machine in a "stand-by" condition, and the increment required for work. Men in industry require, in addition to their basal need, from 600 to 2,500 calories to do their eight hours' work (125 to 300 calories per hour). This means that the increment in energy exchange during work is usually not more than four times the BMR. Since a healthy man is actually capable of increasing his energy exchange ten times for short periods and five times for an eight-hour day, there is a large "factor of safety" in present-day industrial work. This means that there is a reserve of energy for brief emergencies. To an engineer the foregoing facts are elementary, and yet their vital significance in present world economy is not properly appreciated. Nearly all of the old world's population is on short rations. While many underfed people are well as judged by clinical physical examination, they are incapable of doing a full day's work: this has been demonstrated clearly in the Ruhr coal mines. The Ruhr miner in normal times produced two tons of coal and consumed 4,200 calories: 1,800 for basal needs plus 2,400 for work. Cutting his ration to 3,000 calories during the war led to reducing his coal output by one-half. Expressed in another way, removing from his ration the equivalent of one pound of bread (1,200 calories) would cut his daily coal output by 2,000 pounds. Engineers should appreciate that, when their professional duties call them to underfed areas, either their staff and working force must be adequately fed, or estimates of working capacity must be scaled down to correspond to the worker's food intake. What actually happens is low productivity while on the job and absenteeism. Food supplements are not a simple solution; many workmen quite naturally will take extras home to their families.

Nutrition. In the U.S.A. in recent years and even in the present era of high prices, caloric deficiency is not of major importance. Emphasis by nutritionists has been on accessories, particularly vitamins. While there is some evidence for gain in the output of industrial workers when they are given vitamin supplements, this is distinctly a second-order phenomenon. Most enthusiastic vitamin propagandists have been taken aback by the virtual absence of clear-cut evidence of vitamin deficiency diseases in England, where intake of vitamins is far lower than in the U.S.A. Their recommended standard for ascorbic acid, for example, is one-third of ours, and yet reports of scurvy are as rare in England as in the United States.

War experience has emphasized that attention to food habits and to the acceptability of food pays off in large scale feeding. Many large industrial plants have adopted the policy of providing palatable, inexpensive food in their own cafeterias; others contract for such service with organizations expert in this field. The consensus of opinion is that, whichever method is adopted, the investment is sound; that worker appreciation returns dividends in lower personnel turnover and absenteeism and in increased productivity. Unfortunately, there is no record of a well-controlled experiment with facts and figures, a sad commentary on the rudimentary state of our knowledge of industrial physiology.

Human Engineering. This is a new field which physiologists and psychologists are cultivating, or, perhaps better, preparing for cultivation. Rapid strides were made during the war, notably in the Army Air Force, whose Aero Medical Laboratory at Wright Field called in anthropologists, physiologists, and engineers to redesign their machines to fit the men who must fight in them. The pressure was great to accomplish this job successfully, since the lives of young men were at stake. In the past, industry has given only minor attention to human capacities and limitations in designing their machines and tools. To bring this to the industrial level, what rational basis is there for our present size of bricks, of cinder blocks, of other building units? How much attention has been given to the physiology and anatomy of operators by the designers of textile machinery? And what facts have we regarding the relation of floor construction to the fatigue of workers? Such questions are engaging the attention of some physiologists and psychologists and have encouraged some universities to organize Departments of Human Engineering. Their research programs promise to contribute to the well-being of the worker and the satisfaction he takes in his job.

The Physical Environment. Comprised in the Physical environment are such diverse factors as temperature, humidity, air movement, ventilation, dust, odours, toxic agents, illumination, colour, vibration of floors, noise, and music. Some of these are clearly of concern to physiologists. Dust and toxic agents are major responsibilities of the industrial hygienists, of schools of public health, of toxicologists. For example, three-quarters of the papers presented before the annual meeting of the American Industrial Hygiene Association may lie within this area. Illumination, colour, vibration (using this word in the popular sense), noise, and music have received little attention from industrial hygienists or from physiologists. Some psychologists have become experts on these subjects, since they do concern the special senses. But, generally speaking, research on illumination, colour, and music for the man and woman at work has been done by the engineer whose sponsor has fixtures or glass or paint or music for sale. It is granted that factories were once dark, dull, and noisy; that modern factories are bright and cheerful. Yet there has been little basic research on the amount and type of illumination, the colour, and the music best suited to the various conditions of

* The word *calorie* as used here is the kilogram-calorie.

modern industry Here again is a golden opportunity for unbiased experimentation on the relation of environmental conditions to the happiness and productivity of the man at work.

The Fatigue Problem. This is one of the most baffling subjects for the industrial physiologist. There is no question but that men become tired during their day's work and it is equally certain that, in some cases, their fatigue can be related to the amount of physical work done or to one or more distressing features of their environment—such as heat, humidity, noise, or vibration. But other workers who are subject to none of these stresses may also be fatigued by their day's activity. Their fatigue may be the outcome of a multiplicity of stresses, some of which may be quite apart from their specific job but which may be entirely at the emotional level. In other cases, their fatigue may arise from the lack of any pleasant emotional stimuli; i.e. from boredom. Here the physiologist must turn to the psychologist, the plant physician, or the personal councillor. Whatever his title may be, he plays the role of the family physician or the priest in that he (or she) has a sympathetic ear and an understanding of human relations. This is a field in which Elton Mayo and his associates, formerly of the Harvard Graduate School of Business Administration, did pioneer research at the Hawthorne plant of the Western Electric Company beginning about twenty years ago.

Future of Research in Industrial Physiology. Having pointed out some of the contributions of physiology to industry and indicated new approaches of this science to man at work, it seems clear that much remains to be done. The question is, where are these investigations to be carried out ? The history of the Fatigue Laboratory at Harvard University illustrates one failure of industrial physiology to "catch on" in a great university. Established in 1927 by Henderson, Mayo, Donham, Edsall, and Lowell, its doors were closed twenty years later for lack of support, despite notable contributions to our understanding of the physiology of man at work.

The history of similar undertakings in other nations is brief. In England, a useful function has been served by the Industrial Health Research Board. In the USSR several physiological laboratories were established in large manufacturing plants about fifteen years ago, but records of scientific research in such laboratories remain inaccessible. Ambitious projects for research institutes in industrial physiology launched in Japan and Italy were interrupted by their plunge into war. Germany was in a leading position because of her Kaiser Wilhelm Institute for Work Physiology at Dortmund. A large and active staff worked there until the middle of the war, when evacuation was necessary. *Arbeitsphysiologie*, a journal of work physiology international in scope, was published there until 1944.

The future of such research in the U.S.A. seems brighter in State-supported universities and in Government laboratories such as those of the National Institute of Health. Notable among university research centres is the "Laboratory of Physiological Hygiene" at the University of Minnesota. Dr. Keys spent the years of 1934-36 at the Fatigue Laboratory following a research fellowship at Cambridge, England, and at Copenhagen. He has built up an effective organization, with a competent staff alert to the possibilities of research in industrial physiology. This laboratory is still expanding in facilities and personnel; it deserves stable and adequate support, assured for the indefinite future.

Finally, a move by the American Physiological Society to establish a *Journal of Applied Physiology* merits the attention of the readers of *The Engineers' Digest*. Preliminary plans have crystallized and, if ratified by the Society meeting, the journal will be launched in July, 1948. This will provide a medium where physiological research on man at work can be published. It will be useful to engineers who wish to keep up-to-date on such research.

U. S. A.

Disk Stresses

By V. G. GUINS and G. H. HEISER. (From *Machine Design*, Vol. 20, No. 2, February, 1948, pp. 144-148, 188, and 190, 2 illustrations.)

NOMENCLATURE.

A, B	=	Constants
p	=	Radial stress
q	=	Tangential stress
r	=	Radius
ρ	=	Mass density
ω	=	Angular velocity
μ	=	Poisson's ratio
h	=	Disk thickness
k	=	$-p/q$ for static disk at the bore
δ	=	Fit of disk on shaft
D	=	Nominal disk bore diameter
ΔD	=	Increase of bore diameter due to centrifugal force
E	=	Modulus of elasticity

THERE are various methods for determining the stresses due to rotation in disks. The sum and difference method has been described[1] and used widely. Its main shortcoming is that the method gives the disk stresses due to rotation only. To determine the bore stresses due to a shrink or press fit some other just as laborious method must be used.

Where a hyperbolic profile method can be used it is usually confused by a hub and the fact that it is not practical nor economical to machine the profile to the shape used in the calculation.

The method to be described was partly suggested by a recent paper by K. E. Bisshopp[2], and the mode of attack is similar to that of the sum and difference curves. The "dead" load of the blades and rim is calculated and the radial stress is determined at some line above which there is no tangential load-carrying ability of the disk. Below this line the disk is divided into several convenient parallel-sided disks which approximate the disk's shape.

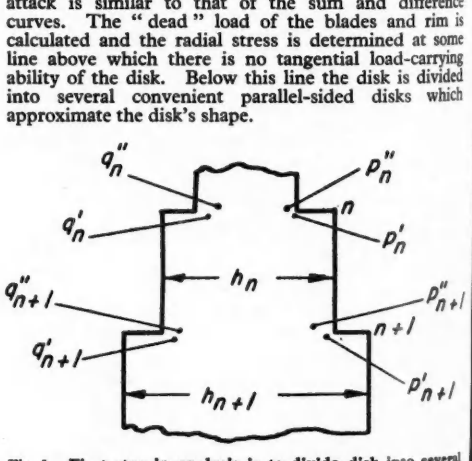

Fig. 1. First step in analysis is to divide disk into several parallel-sided disks approximating actual shape.

Referring to the Nomenclature for a parallel-sided disk, the well known general formulae can be written :

$$p = A - \frac{B}{r^2} - \frac{1}{8}(3 + \mu)\rho\omega^2 r^2$$

$$q = A + \frac{B}{r^2} - \frac{1}{8}(1 + 3\mu)\rho\omega^2 r^2$$

Considering the nth section as shown on Fig. 1 and letting p_n' and q_n' represent the radial and tangential stress just at the start of the section of h_n width,

$$p_n' = A - \frac{B}{r_n^2} - \frac{1}{8}(3 + \mu)\rho\omega^2 r_n^2$$

and

$$q_n' = A + \frac{B}{r_n^2} - \frac{1}{8}(1 + 3\mu)\rho\omega^2 r_n^2$$

Adding the foregoing stresses,

$$p_n' + q_n' = 2A - \frac{1}{2}(1 + \mu)\rho\omega^2 r_n^2 \quad .. \quad (1)$$

and by subtracting them,

$$p_n' - q_n' = -\frac{2B}{r_n^2} - \frac{1}{4}(1 - \mu)\rho\omega^2 r_n^2 \quad .. \quad (2)$$

Just at the end of the section h_n width,

$$p_{n+1}'' = A - \frac{B}{r_{n+1}^2} - \frac{1}{8}(3 + \mu)\rho\omega^2 r_{n+1}^2 \quad (3)$$

$$q_{n+1}'' = A + \frac{B}{r_{n+1}^2} - \frac{1}{8}(1 + 3\mu)\rho\omega^2 r_{n+1}^2 \quad (4)$$

the sum and difference of eqs. (3) and (4) are

$$p_{n+1}'' + q_{n+1}'' = 2A - \frac{1}{2}(1 + \mu)\rho\omega^2 r_{n+1}^2 \quad (5)$$

$$p_{n+1}'' - q_{n+1}'' = -\frac{2B}{r_{n+1}^2} - \frac{1}{4}(1 - \mu)\rho\omega^2 r_{n+1}^2 \quad (6)$$

Subtracting eq. (1) from eq. (5) and rearranging,

$$p_{n+1}'' + q_{n+1}'' = p' + q_n' + \frac{1}{2}(1+\mu)\rho\omega^2(r_n^2 - r_{n+1}^2) \quad (7)$$

If eq. (2) is multiplied by $\frac{r_n^2}{r_{n+1}^2}$, subtracted from eq. (6) and rearranged,

$$p_{n+1}'' - q_{n+1}'' = (p_n' - q_n')\frac{r_n^2}{r_{n+1}^2} + \frac{1}{4}(1 - \mu)\rho\omega^2(r_n^2 - r_{n+1}^2) \times \left(1 + \frac{r_n^2}{r_{n+1}^2}\right) \quad (8)$$

Eqs. (7) and (8) will give the radial and tangential stress at the inner radius of the section of h_n width provided the radial and tangential stress at the outer radius of this section are known. The transition from section to section is made in the conventional manner. The radial stress is assumed to vary inversely as the thickness at the transition, hence

$$p_{n+1}' = \frac{h_n}{h_{n+1}} \times p_{n+1}'' \quad .. \quad .. \quad (9)$$

The change in tangential stress at the transition from one thickness to another is then equal to Poisson's ratio times the change in the radial stress ; then

$$q_{n+1}' = q_{n+1}'' - \mu\left(\frac{h_{n+1}}{h_n} - 1\right)p_{n+1}' \quad .. \quad (10)$$

Eqs. (7), (8) (9) and (10) now give enough tools to calculate the stress distribution in a disk. The tables in which a sample calculation is listed can be readily filled out. In the solution for the stress distribution in a disk with a free bore two conditions are known : One is the radial stress at the outer diameter while the other is the radial stress at the bore. The latter, of course, is zero, while the stress at the outer radius is the stress due to the radial load above the diameter where the calculation begins. For a solution these end conditions must be met.

In the use of the sum and difference curves the

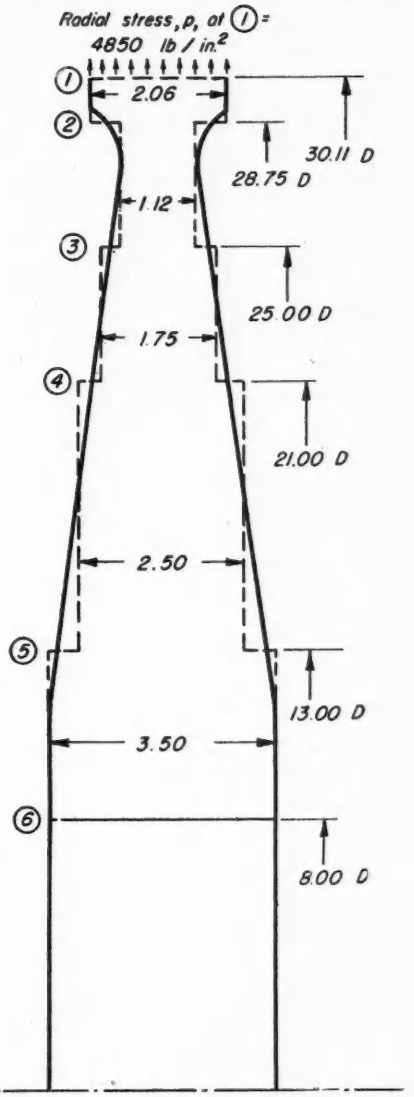

Fig. 2. Actual and approximate shape of alloy steel disk which serves as example illustrating the procedure.

TABLE I. EVALUATION OF CONSTANTS FOR DISK.

$N = 3987$ r.p.m. $\omega = 418$ rad. per sec. $[A] = \frac{1}{8}\rho(1+\mu)\omega^2 = 83{\cdot}3$

$\mu = 0{\cdot}30$ $\rho = 7{\cdot}33 \times 10^{-4}$ $\omega^2 = 17{\cdot}5 \times 10^4$. $[B] = \frac{1}{8}\rho(1-\mu)\omega^2 = 22{\cdot}4$

	[C]	[D]	[E]	[F]	[G]	[H]	[J]	[K]	[L]	[M]
n	r_n	h_n	$\frac{h_{n+1}}{h_n}$	$\mu\left(\frac{h_{n+1}}{h_n}-1\right)$	r_n^2	$r_n^2 - r_{n+1}^2$	$\frac{r_n^2}{r_{n+1}^2}$	$\frac{[H]\times}{\{1+[J]\}}$	[A] × [H]	[B] × [K]
1	15·06				227	20·5	1·10	43·0	1710	963
2	14·375	2·06	0·545	−0·1365	206·5	50·3	1·32	116·6	4190	2610
3	12·50	1·125	1·555	0·1665	156·2	45·9	1·417	111·3	3830	2490
4	10·50	1·75	1·428	0·1283	110·3	68·1	2·62	246·0	5670	5520
5	6·50	2·50	1·400	0·120	42·2	26·2	2·63	95·1	2180	2130
6	4·00	3·50			16·0					
7										
8										

usual practice is to assume a tangential stress at the outer diameter and work inward to the bore. If the radial stress at the bore is not zero then another assumption is made and the curves gone through again. Having two trials, a third trial should be final since the radial stress at the bore will vary as a straight-line function with the tangential stress at the outer diameter.

In the method described in this article the tangential stress at the outside is assumed (10,000 lb./sq. in. in the example) just as in the sum and difference curve method but, instead of using the curves, the stress condition at the bore is found by filling out the table as illustrated in Table II. The example is for the disk shown in Fig. 2, on which the radial stress at the periphery is 4850 lb./sq. in.

First the constants and the values indicated in Table I are listed. The stress condition that exists for Table II at the bore is the bore stress for the assumed tangential stress (10,000 lb./sq. in.) at the outside diameter. Whatever the radial stress is at the bore, a radial stress of opposite sign but the same numerical value could be superimposed and the radial stress at the bore would become zero and the stress distribution throughout the disk would change. Table III, is then filled out, using the same eqs. (7), (8), (9) and (10) except that all terms that are affected by centrifugal force are neglected. The same assumed tangential stress at the outside diameter can be used for convenience.

The superposition involving Tables II and III to secure zero radial stress at the bore can be justified as follows. The stress condition indicated in Table II is due to centrifugal force and a radial pull at the bore which turns out to be 5110 lb./sq. in. The conditions indicated in Table III are due only to a radial pressure at the bore of 51,225 lb./sq. in. Then superposition of the proper part of the stresses in Table III on Table II to eliminate the radial pull of 5110 lb./sq. in. will involve no change in the centrifugal loading.

Summarizing what Table II shows it is evident that this is the stress distribution of the disk under the conditions assumed, namely a radial stress of 4850 lb./sq. in. and a tangential stress of 10,000 lb./sq. in. at the outer diameter. The radial stress 5110 lb./sq. in. at the bore indicates a pull of that amount is necessary to give the outside conditions. In Table III the stress distribution is that of a stationary disk with a radial pressure of 51,200 lb./sq. in. at the bore.

Having correctly filled out Tables II and III, the bore stress can quickly be determined. To have zero radial stress at the bore in Table II a radial pressure of 5110 lb./sq. in. must be superposed on the bore pull of 5110 lb./sq. in., or 5110/51,225 = 0·0998 of the radial pressure indicated at the bore in Table III. Then the stress distribution of Table II changes likewise throughout the disk and to find the correct values it is only necessary to multiply the values in Table III by 0·0998 and add to the values in Table II, taking due account of the signs. Thus the free bore stress becomes 18,630 + 0·0998 × 70,800 = 25,700 lb./sq. in. It is seen that the assumed tangential stress at outer diameter in Table II should have been 10,000 + 0·0998 × 10,000 = 11,000 lb./sq. in.

For a check on the arithmetic involved in the tables, Table IV can be filled out, using the new tangential stress at the outside of 11,000 lb./sq. in. The radial stress at the bore now comes out closely zero and there is the same tangential bore stress as determined in the previous paragraph. A radial bore stress checking within plus or minus 100 lb./sq. in. will indicate sufficient accuracy.

TABLE II. TRIAL CONDITIONS, FOR PERIPHERAL STRESSES 4850 LB./SQ. IN. RADIAL AND 10,000 LB./SQ. IN. TANGENTIAL, AT WORKING SPEED.

	[N]	[P]	[Q]	[R]	[S]	[T]	[U]	[V]
n	$\frac{p_n' [U]}{[E]}$	$\frac{q_n' [V]}{[F]\times[N]}$	$p_n' + q_n'$ / [N] + [P]	$p_n' - q_n'$ / [N] − [P]	$p_n'' + q_n''$ / [Q] + [L]	$p_n'' - q_n''$ / [R] × [J] + [M]	$\frac{p_n''}{[S]+[T]}{2}$	$\frac{q_n''}{[S]-[T]}{2}$
1	4850	10000	14850	−5150	16560	−4700		
2	10900	12120	23020	−1220	27210	1000	5930	10630
3	9070	11600	20670	−2530	24500	−1100	14105	13105
4	8200	11750	19950	−3550	25620	−3780	11700	12800
5	7800	13760	21560	−5960	23740	−13520	10920	14700
6							5110	18630
7								
8								

TABLE III. CONDITIONS FOR PERIPHERAL TANGENTIAL STRESS OF 10,000 LB./SQ. IN. AT ZERO SPEED.

	[N]	[P]	[Q]	[R]	[S']	[T']	[U]	[V]
n	$\frac{p_n' [U]}{[E]}$	$\frac{q_n' [V]}{[F] \times [N]}$	$\frac{p_n' + q_n'}{[N] + [P]}$	$\frac{p_n' - q_n'}{[N] - [P]}$	$\frac{p_n'' + q_n''}{[Q]}$	$\frac{p_n'' - q_n''}{[R] \times [J]}$	$\frac{p_n''}{\frac{[S'] + [T']}{2}}$	$\frac{q_n''}{\frac{[S'] - [T']}{2}}$
1	0	10000	10000	—10000	10000	—11000		
2	—918	10380	9460	—11300	9460	14900	—500	10500
3	—1750	12470	10720	—14220	10720	—20150	—2720	12180
4	—3300	15860	12560	—19160	12560	—50200	—4715	15435
5	—13440	32990	19550	—46430	19550	—122000	—18820	31380
6							—51225	70775
7								
8	$k = -\frac{-51225}{70775} = 0{\cdot}724$							

The results of Table III are also useful in that the ratio between the radial and tangential stress is readily available for the static condition. This ratio, k, is necessary to find the tangential and radial bore stresses due to a given press or shrink fit, δ, between the disk and the shaft. To solve for these stresses at the bore the known relations can be written :

$$\left.\begin{aligned} k &= -\frac{p}{q} \\ p &= -kq \end{aligned}\right\} \quad .. \quad .. \quad .. \quad (11)$$

It is convenient to inject the minus sign since p, the radial stress, is a compressive stress under the conditions described and k then will be a positive number. Further, from Hooke's law, at the bore,

$$\frac{\delta}{D} = \frac{1}{E}(q - \mu p) \quad .. \quad .. \quad (12)$$

Combining eqs. (11) and (12),

$$q = \frac{\delta}{D} \times \frac{E}{1 + \mu k} \quad .. \quad .. \quad (13)$$

Eqs. (11) and (13) give the radial and tangential bore stresses due to the fit, δ, for an inelastic shaft. This assumes all the fit goes into deformation of the disk. This is not quite the case since the shaft will be subjected also to radial pressure and will undergo deformation.

If the shaft is assumed not to protrude beyond the hub of the disk, the shaft deformation can be written

$$\delta_s' = -\frac{Dp}{E}(1 - \mu) \quad .. \quad .. \quad (14)$$

The increase in bore diameter of the disk is given by

$$\delta_D = \frac{D}{E} \times q(1 + \mu k) \quad .. \quad (15)$$

Then, substituting $p = -kq$ into eq. (14) and adding eqs. (14) and (15), since $\delta = \delta_S + \delta_D$, and setting equal to the total shrink fit, the equation for tangential bore stress is

$$q = \frac{\delta}{D} \times \frac{E}{1 + k} \quad .. \quad .. \quad (16)$$

After the tangential bore stress due to the shrink fit is determined, the radial bore stress is found from the relation $p = -kq$.

To illustrate the particular example shown in this article let it be assumed that the fit, δ, of the disk on the shaft is 0·022-inch. For a stationary disk, and considering the shaft as rigid, the tangential bore stress, q from eq. (13) is

$$q = \frac{0{\cdot}022}{8{\cdot}00} \times \frac{29 \times 10^6}{1 + 0{\cdot}3 \times 0{\cdot}724} = 65{\cdot}400 \text{ lb./sq. in.}$$

Then from eq. (11) the radial pressure at the bore is

$$p = -0{\cdot}724 \times 65{,}400 = -47{,}300 \text{ lb./sq. in.}$$

Again for a stationary disk, but considering an elastic shaft, the tangential bore stress from eq. (16) is

TABLE IV. FINAL CHECK OF ACTUAL STRESSES IN DISK.

	[N]	[P]	[Q]	[R]	[S]	[T]	[U]	[V]
n	$\frac{p_n' [U]}{[E]}$	$\frac{q_n' [V]}{[F] \times [N]}$	$\frac{p_n' + q_n'}{[N] + [P]}$	$\frac{p_n' - q_n'}{[N] - [P]}$	$\frac{p_n'' + q_n''}{[Q] + [L]}$	$\frac{p_n'' - q_n''}{[P] \times [J] + [M]}$	$\frac{p_n''}{\frac{[S] + [T]}{2}}$	$\frac{q_n''}{\frac{[S] - [T]}{2}}$
1	4850	11000	15850	—6150	17560	—5800		
2	10800	13155	23955	—2355	28145	—500	5880	11680
3	8900	12840	21740	—3940	25570	—3100	13820	14320
4	7870	13325	21200	—5450	26870	—8780	11235	14335
5	6460	17050	23510	—10590	25690	—25770	9045	17825
6							—40	25730
7								
8								

$$q = \frac{0.022}{8.00} \times \frac{29 \times 10^6}{1 + 0.724} = 46{,}300 \text{ lb./sq. in.}$$

also, from eq. (11) the radial pressure at the bore is

$$p = -0.724 \times 46{,}300 = -33{,}500 \text{ lb./sq. in.}$$

For the case of the rotating disk the stress conditions can be found from the superposition of those due to fit and those due to centrifugal force. For the speed used to illustrate the tables (3987 r.p.m.) the free bore stress is 25,700 lb./sq. in. Following Hooke's law, the bore elongation, ΔD, is

$$\Delta D = \frac{25{,}700 \times 8.00}{29 \times 10^6} = 0.0071 \text{ in.}$$

This elongation relieves the 0·022 stationary fit so that at 3987 r.p.m. the effective fit is 0·022 — 0·0071 = = 0·0149 inch. Then, for illustration using the case of the rigid shaft, the tangential bore stress is the sum of the tangential stresses due to the centrifugal force and the shrink fit. This stress is then

$$25{,}700 + \frac{0.0149}{0.022} \times 65{,}400 = 70{,}000 \text{ lb./sq. in.}$$

under these same conditions then the radial stress is

$$-\frac{0.0149}{0.022} \times 47{,}300 = -32{,}000 \text{ lb./sq. in.}$$

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RUSSIA

Friction and Wear of Metals in the Presence of Liquid Gases

By P. I. RIUMIN and YU. N. RIABININ. (From *Kislorod*, No. 4, 1946, pp. 35-41, 5 illustrations.)

FOR the construction of a plunger-type liquid oxygen de-compressor the authors required to know the coefficients of friction and the resistance to wear of various metals in this medium to enable them to make a suitable choice of materials. They found no published data on the subject and as the factors determining the friction properties of surfaces are too numerous, complicated, and uncertain to permit an intelligent guess it was decided to choose from results obtained experimentally. The experiments are not claimed to have been exhaustive yet they may be of practical interest.

The experiments were conducted by pressing the specimens representing one rubbing surface against a rotating disk which represented the other rubbing surface. The peripheral speed was about 3 ft./sec. and the "distance" traversed in each wear test was 3,600 feet after which the loss of metal was determined by weighing. Tests were made both in liquid oxygen and in oxygen gas very near its boiling point. The pressure between the rubbing surfaces could be varied from 45 to 450 lb./sq. in. but the main wear tests were made with a pressure of 280 lb./sq. in.

The first series of experiments was tests of various materials in liquid oxygen rubbing against a disk of stainless steel. The weight of material worn away (in milligrams) varied from 3·9 to 6·9 for bronzes, 16 to 146 for white metal and 78 for brass. In spite of the apparent superiority of bronzes, they were not considered to have proved very satisfactory because they tended to smear on to the steel, thus creating a danger of seizure.

The second series of experiments was to ascertain the extent to which oxidation plays a part in the process of wear. Comparative figures were taken for liquid oxygen and liquid nitrogen, and it was found that wear in oxygen was 1·5 to 2·8 times greater when two bronzes were tested against white-metal.

In the third series of experiments it was desired to ascertain whether the liquid gas had any lubricating properties and comparative figures were taken at varying speeds in liquid and gaseous oxygen, both near the boiling point, i.e. as nearly as practicable at the same temperature. It was found that the liquid does act as a lubricant, because the coefficient of friction figures in gas were both 50 to 80 per cent higher than in liquid, and noticeably less consistent.

The fourth tests were run to ascertain the effect of variation in wear and coefficient of friction in liquid oxygen with varying pressures. The speed was kept constant at 2·9 ft./sec. and the pressure was varied through the maximum range stated above.

Wear was found to be almost proportional to pressure whilst the coefficients of friction were as follows :—

Constant at 0·41 for Caro Bronze ; falling with increase of pressure from 0·44 to 0·26 for Babbitt metal and falling from 0·55 to 0·48 for specimens of nitrided steel against a disk of the same material. This last pair of materials was tested in an effort to reduce wear which was found to be too high when using anti-friction materials. And indeed, although it showed a rather high coefficient of friction, wear was reduced to one 3rd to one 5th compared with that of bronze and to one 16th compared with one of the more successful white metals. Once again comparative tests in nitrogen showed a greatly reduced wear (2-4 times less).

For the final series of tests—to ascertain the most durable material—the duration of each test was increased and the shape of the specimen altered so that the figures are not comparable with those of the first series. Some of the more interesting figures are given below :—

MATERIAL.	WEAR. milligrams
Hardened tool steel against hardened tool steel	0·55
Hardened tool steel against nitrided steel	0·70
Case hardened stainless steel against case hardened stainless steel	0·80
Case hardened stainless steel against case hardened stainless steel (another quality)	1·1
Case hardened Ni-Steel against nitrided steel	2·4
Stainless steel against case hardened stainless steel	12·1
White-metal against stainless steel ..	15·4

Substitute Materials for Overhead Wires for Tramway and Trolleybus Services

By L. ALBERT. (From *Revue de l'Aluminium*, Vol. 25, No. 140, January, 1948, pp. 3-12, 7 illustrations.)

THE copper shortage in France led to the development of substitutes for trolley wire materials, consisting generally of a "contact" wire, or facing, of steel, rigidly or loosely connected with the aluminium or aluminium alloy cable which carries the current. Previous experience[1, 2] had shown that substitutes can be satisfactory, that the contact portion should not be made from Al or Al-alloy, that separate contact and current carrying cables are unsightly, present difficult connexion problems and should not be used. Composite Al-steel cables seemed to offer the best prospects. In the meantime, about 61 miles of composite Al-steel cables have been put into service, of which 53 miles are of the "Alutroll" type (Fig. 1), and 39 miles of "Almelec" alloy (up to 0·35 per cent Fe, 0·6 per cent Si, and 0·7 per cent Mg) without steel, and about 3 miles of steel cable. It is important to note first that a large radius of curvature at the supports results in less cable wear and that profiles and materials should be chosen correspondingly. Substitute materials compare favourably with copper cables in this respect.

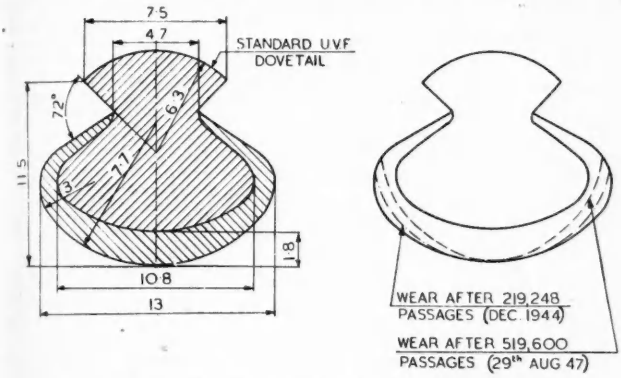

Fig. 1. **"Alutroll" type cable used for tramways at Bordeaux.**
At left, new cable, laid December 3, 1942 (cross-section: Al 70 mm², steel 30 mm²), at right, wear at different stages. Collector shoes of soft cast iron, contact pressure 6 kg, monthly graphitization.

"Alutroll" cables, Fig. 1. Authorities at Lille are disappointed with results, probably because collector rolls were used instead of specially adapted collector shoes made of soft cast iron. Experience at Nice has proved entirely satisfactory as regards wear, erection, and joining first by welding and then by notch glands, and the system is being expanded. Bordeaux and Lyons are satisfied except for certain difficulties concerned with the quality of the material and with junctions. However, improved materials and a new superior gland connexion have been developed by the cable makers. At Bordeaux, the section shown in Fig. 1 has been replaced by a heavier section with a thicker steel facing so that service duration is considerably improved. Monthly graphite lubrication is resorted to, but is hardly necessary. General experience shows that "Alutroll" cables are satisfactory and last longer than copper cables, but that repairs are more complicated and difficult. Technical specifications must be carefully worked out and acceptance conditions for the material rigorously enforced. Several disadvantages remain to be rectified. However, other steel-flanged profiles seem to be generally preferable to the steel faced "Alutroll" type cable.

"Almelec" cables, Fig. 2. Experience at Le Mans, Limoges, and Nice shows rapid wear, need of frequent lubrication, although on the whole, behaviour is satisfactory. This type is applicable in exceptional cases of infrequent traffic and in non-corrosive atmospheres, but is not recommended, at present, for ordinary use.

Compound Al-steel cables other than "Alutroll" type. Fig. 3 shows the profile chosen for an experimental track in Paris; similar profiles are used in Lyons and Rouen. The dovetailing of the two parts gives a rigid joint. At junctions, the aluminium is removed for the length of the gland and the gland is slipped over the steel dovetail. Electrical connexion is made by a conducting bridge between the aluminium ends. The junction is thus easily adjustable and erection is fairly

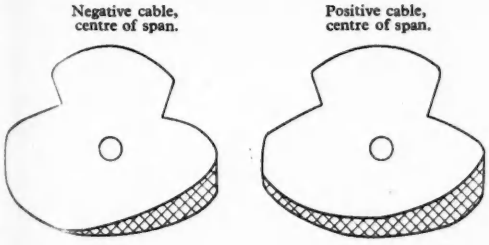

Fig. 2. **Wear, after 80,000 passages, of "Almelec" type cable used at Le Mans.**
(122 mm² cross-section).

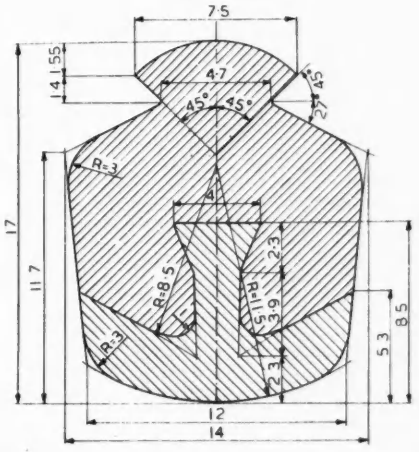

Fig. 3. **Composite Al-steel cable on test at the Metropolitan Railway at Paris.**
(Cross-section: Al 125 mm², steel 53 mm².)

simple. Owing to difference in thermal expansion and the relatively large distance between the centres of gravity of the parts, the cables at Lyons showed warping during the first cold season. These and other possible defects of the profile are still to be remedied. The installation at Lyons has been in use since 1941 without any change of cable. The experimental track at Paris has given every satisfaction. Graphite lubrication adheres better than on copper cables and wear after 77,000 passages of steel collectors is hardly noticeable. Tests will be further followed up.

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NORWAY

Comparison of Fluorescent Tubes with Incandescent Filament Lamps from the Aspect of Flicker due to Voltage Fluctuations

By H. AAS. (From *Elektroteknisk Tidskrift*, Vol. 61, No. 4, February, 1948, pp. 43-56 ; 18 illustrations.)

THE employment of welding sets may cause considerable voltage fluctuations in networks. In view of the recent growth in the application of fluorescent tubes it is of interest to ascertain their sensitivity to voltage fluctuations. A series of tests was carried out at the laboratory of the Oslo Electricity Company in order to clarify this problem.

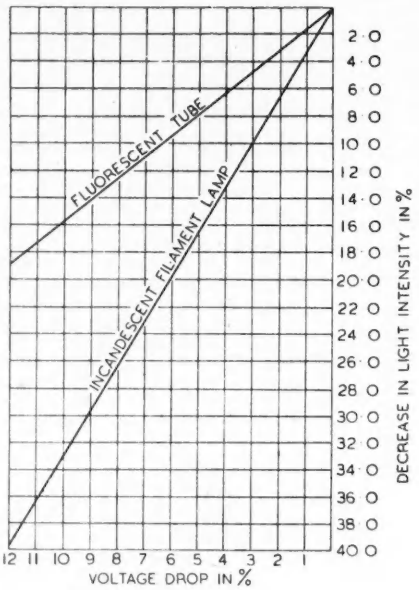


Fig. 1.

The influence of voltage drop upon light emission of an incandescent filament lamp and a fluorescent tube respectively is shown in Fig. 1, the normal voltage being 230 volts. The voltage fluctuations were effected gradually so as to permit the establishment of stable light emission at every stage. It will be seen that for identical voltage drops the decline in the light emission of the fluorescent tube is smaller than that of the incandescent filament lamp. Thus, for a voltage drop of five per cent, the light intensity of the incandescent filament lamp drops by 16·5 per cent as compared with only 8 per cent in the case of the fluorescent tube. With sudden voltage variations the hot filament of the incandescent filament lamp shows a certain thermal inertia which produces a time lag. The fluorescent matter of the fluorescent tube also possesses a certain inertia, but this is much less than that of the incandescent filament and the question arises as to what extent fluorescent tubes are adversely affected by the sudden voltage fluctuations caused by the starting of motors, the operation of welding sets, etc. A special test circuit, outlined in Fig. 2, was devised for the purpose of this investigation. Here the light emission is measured by means of a photoelectric cell F of very small inertia. This photoelectric cell is supplied with a constant direct current voltage of 90 volts, which causes a current of 12 μA to flow if the light intensity is 10^{-2} lumen per sq. cm. The photoelectric cell characteristic is practically a straight-line function of the light intensity up to 18 μA, and the cell follows light intensity variations with frequencies of up to 100,000 cycles per second. The maximum colour sensitivity of the cell lies more

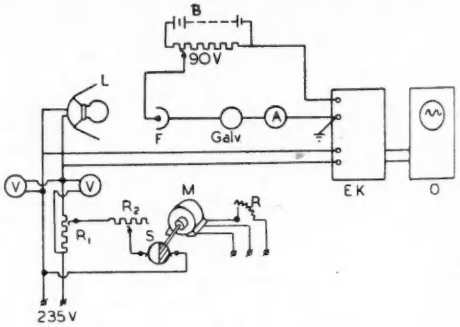


Fig. 2.

B = Battery ; M = Motor ;
E.K. = Amplifier ; O = Cathode-ray oscillograph ;
L = Photoelectric cell ; S = Interruptor.

towards red (at about 7,200 Ångström) than that of the human eye. But this effect was found to be too small to justify the use of a special filter. The photoelectric cell current was registered by the cathode-ray oscillograph O.

A detailed account of the comparative tests conducted is given in the original article, and the following conclusions are arrived at :—With a given periodically occurring voltage drop, the fluorescent tube suffers a smaller objective variation in light intensity than the incandescent filament lamp if the period of voltage drop is larger than the time constant of the latter. This will generally apply to periods of voltage drop in excess of one to two cycles. With a given periodically occurring voltage drop, however, the subjective flicker effect is more favourable for the fluorescent tube than for the incandescent filament lamp if the duration of the voltage drop is longer than two times the time constant of the incandescent filament lamp, or generally two to four cycles. Voltage depression periods of about three cycles therefore represent a dividing line. With shorter durations, the fluorescent tubes become rapidly inferior, whilst for longer periods they prove superior to incandescent filament lamps.

The Influence of Self-Regulation and of the Damping Period on the WR² value of Hydro-Electric Power Plant

By TH. STEIN*. (From *Schweizerische Bauzeitung*, Vol. 65, Nos. 39, 40 and 41, September/October, 1947, pp. 531-535, 543-547, and 564-568, respectively, 10 illustrations.)

GOVERNING AN OBSTACLE TO PROGRESS.

THE development of water turbines has now reached a stage where any possible economy in weight is no longer dictated by design but by the exigencies of governing.

This article is not concerned with the question of providing a flywheel, visible or otherwise, in the generating plant, but with the conditions and means by which satisfactory governing may be achieved by the mechanical inertia obtained by design.

The evolution of water turbine design during the last 100 years is shown in Fig. 1. Efficiency has improved from 60 - 70 per cent to more than 90 per cent and there is now no hope of increasing efficiency under full load to an extent which would affect economy.

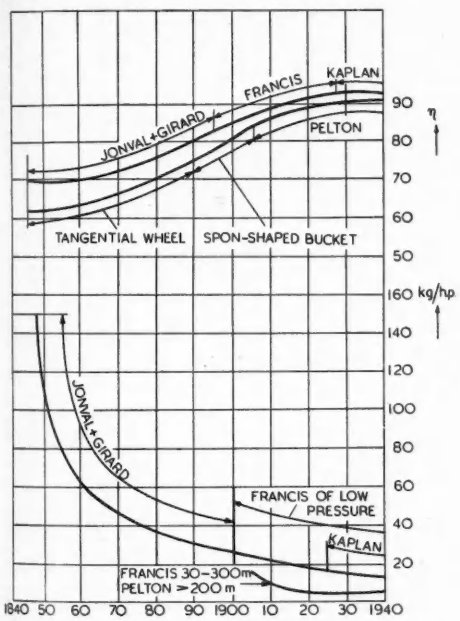


Fig. 1. **Improvement in efficiency and specific weight of water turbines during a century shows that further economical development can only be expected with respect to reduced material consumption, kg./h.p.**

However, a reduction in the installation cost of turbines may still be realized by reducing their weight per horse-power. This value has changed from 150 kg/h.p. to 3 kg/h.p. for turbines of high or medium pressures, and to 12 kg/h.p. for low pressures. Designers have recently striven for a progressive increase in the specific speed n_s. A high n_s value has not only the advantage of lower turbine cost, but also leads to a corresponding reduction in the cost of alternators.

The rotating masses, reduced by the increase in the n_s value, have an acceleration period T_a—corresponding to mechanical inertia—which has diminished from more than 10 to less than 6 seconds. (See Table I.)

TABLE I. OLD AND NEW VALUES FOR ACCELERATION PERIODS T_a AND T_t

T_a sec.	T_t sec.	T_a/T_t	Criterion for stability.
Formerly >10	<1	>10	i.e. now five times
Present Day <6	up to 3	<2	more unfavourable

Conversely, the effect of the hydraulic mass has increased with the length of the pressure pipe and the discharge velocity of the water. This has resulted in an increase from the former 1 sec. to 2 or 3 seconds, as the characteristic acceleration period of the water column T_t—corresponding to hydraulic inertia.

The following explains the difficulties encountered in stabilizing a hydro-electric power plant.

The power output variation of a turbine follows the servo-motor movement with a delay which increases with the hydraulic inertia. Conversely, with the reduction in mechanical inertia resulting from the continual increase of the n_s value, the variation in speed follows more and more closely the variation in power. Hunting could therefore be avoided only by a decrease in the speed of the servo-motor piston.

Thus governing stability is determined by the ratio of the mechanical inertia (T_a) to the hydraulic inertia (T_t) (see Table I).

The value of this ratio has fallen from a former value of 10 to 2 and it has thus become 5 times more difficult to achieve stability of governing. It therefore appears that the stability limits do not permit the reductions in weight provided by an increase in the n_s value.

In seeking to avoid this obstacle a study of all relative factors must be made, in a sufficiently simple and general manner, to allow for direct application of the information obtained.

In practice, theory was used only for calculating the limiting case of governing with undamped oscillations, without reference to the self-regulating effect provided by the variations of frequency in the circuit. The meaning of self-regulation will be explained later.

Then, values were fixed by trial and error for a correction factor but these diverged widely because the effect of self-regulation varies in each case and governor characteristics must be determined according to the damping rate adopted.

The extent to which automatic governing is assisted by self-regulation and to which the demand for more rapid damping renders such governing more difficult will be shown by curves of general validity. These curves allow a large part of the disagreement between theoretical and experimental values to be eliminated. One of the means for ensuring good stability, without the necessity of increasing mechanical inertia, lies in the amplification of the self-regulating effect when it is inadequate, and the extent to which this amplification is required can be deduced directly from the same curves.

In addition to the dash-pot type of governor, in which the function of the dash-pot is to provide a large temporary deviation, other types are used in which the dash-pot is replaced by a device which responds according to the acceleration. Although their design is very different, the regulating process in these two governors is identical, and the delay in response or the sluggishness of governors, is of prime importance.

In a dash-pot type of governor the delay increases with the temporary deviation and with the rigidity of

*Escher Wyss Ltd., Zurich.

the dash-pot, and it is the product of these two values. To clarify rapid manual overriding control, it should be remembered that the servo-motor may follow the dash-pot without any supplementary retardation, and the retardation required for stability is produced entirely within the dash-pot. By directly operating the distributor valve of the servo-motor by remote control, a rapid manual overriding control is provided which is in no way reduced by the delay in the automatic governing process. In the case of the acceleration-type governor, the delay is the product of the operating time and the difference in r.p.m. required to reach the maximum speed of the servo-motor. The operation of dash-pot-type and acceleration-type governors is identical when the delay values are the same and when the period representing the rigidity of the dashpot is the same as that representing the characteristic time of the acceleration-type governor. The stability curves given later are based on these conditions of identity.

SELF-REGULATION.

When the r.p.m., or the frequency increases, more power is used by grid consumers so that the alternator torque increases. At the same time the turbine torque decreases. Thus there is a difference in torque which facilitates governing. If the speed regulator were inoperative, the speed would increase only until the decreasing turbine torque and the increasing resisting torque reached equilibrium, and this self-regulating effect is provided without artificial governing but is subject to inadmissible fluctuations in the speed. However, as self-regulation acts in the same sense as the governor, it may offer useful assistance.

TABLE II. ORDER OF MAGNITUDE OF SELF-REGULATION COEFFICIENTS.

	e_{sg}	e_{st}	$e_s = e_{sg} - e_{st}$
TURBINE :			
In general	—	about — 1	—
High specific speed	—	up to — 0·6	—
GENERATOR :			
Grid loading : Motors only (Constant Torque)	0	—	+ 1
Ohmic resistance only with voltage regulator ..	— 1	—	0
Without voltage regulator	+ 1 to + 4	—	2 to 5
Paris Grid[1]	0·5 to 1·5	—	1·5 to 2·5

(1) According to Nasse, written communication.

Self-regulation does not only occur in water turbines or other prime movers. In a steam system, an increase in pressure increases consumption while reducing the discharge from the controlled valve. In 1928 the author explained, why pressure controllers are stable in operation, even without the use of servo-motors, contrary to previous theories. In 1940 the theory based on self-regulation of the constant-speed unit for variable pitch airscrews was established. Self-regulation may be effective in this latter application and for the water turbine, but it is practically inoperative in the case of the steam turbine in spite of the similarity between the two types. This explains the necessity for a preliminary analysis of the effectiveness of self-regulation in water turbines.

The following equation is particularly instructive :—

$$T_a \frac{d\varphi}{dt} = \mu + 1{\cdot}5\, z\, \varphi_h - z_0\, e_s\, \varphi$$

Rotating masses — Servo-motor — Pressure impact — Self-regulation

Where T_a = Characteristic time to accelerate rotating masses (WR^2).
φ = Instantaneous speed deviation.
φ_h = Instantaneous pressure deviation (pressure impact).
μ = Instantaneous servo-motor deviation.
e_s = Self-regulation coefficient.
z_0 = Load factor related to output.
z = Load factor related to water consumed.

With water-turbines, self-regulation does not make a restoring mechanism unnecessary as in the case of pressure-type governors for steam turbines and air-screw governors. It is intended rather to compensate for the unfavourable effect of water inertia. To show that self-regulation is capable of this, imagine the terms in the equation for self-regulation and for pressure impact to be suppressed. The increase in the speed $d\varphi/dt$ is then proportional to the relative motion of the servo-motor μ and increases as the mechanical inertia T_a decreases. Now let us introduce the pressure impact $1{\cdot}5\ z\varphi_h$ without self-regulation. If the increase in r.p.m. $d\varphi/dt$ is to be the same, the mechanical inertia must be increased. Self-regulation, however, acts in opposition to the pressure impact. The term for pressure impact includes a hydraulic load factor z, related to the discharge of water, and that for self-regulation, a load factor z_0 related to the power supplied. As a first approximation these two factors may be assumed to be equal. It follows, therefore, that the self-regulating effect is able to compensate for the pressure impact since the influence of both these factors increases with the load.

If the self-regulation coefficient e_s is large enough, the self-regulating effect is sufficient to compensate the pressure impact completely. In this case the increase in r.p.m. $d\varphi/dt$ is not greater than it would be if the pressure impact were zero. With larger self-regulation coefficients the rotating masses can be progressively reduced, at least in the case of high load factors. The equation shows, moreover, that it would be possible, from the stability aspect, to reduce the rotating masses to zero.

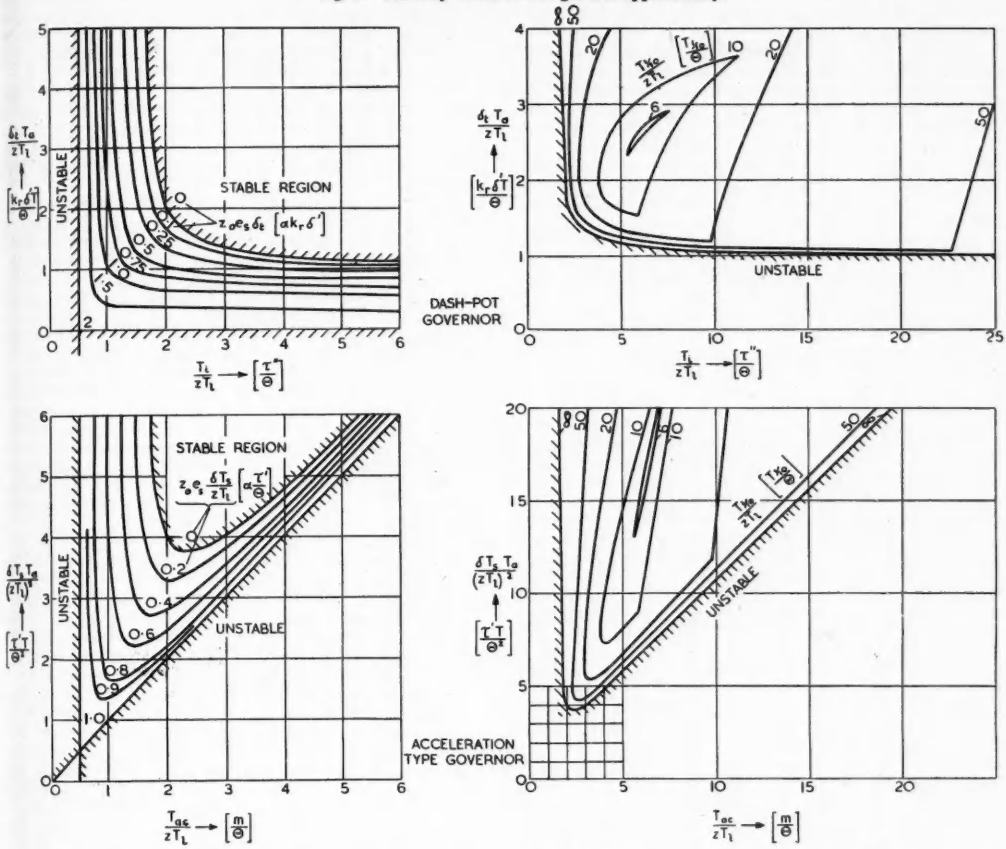
The self-regulation coefficient e_s may be defined by the relative variation in the torque with a variation of 1 per cent of the speed, i.e. of the frequency, Table II. For normal turbines with increase in speed of 100 per cent when racing, the value of the self-regulation coefficient for the turbine is —1. With turbines of high specific speed and with higher racing speeds the value approaches —0·6. The generator then has no self-regulating effect ($z_0 = 0$) if it feeds only constant torque motors. The total self-regulation coefficient, which is the difference between the torque of the generator and that of the turbine is equal to +1.

The case of a purely ohmic load with voltage control is critical. The power is constant and the torque decreases 1 per cent for each percentage variation of the r.p.m. The self-regulation constant is —1 for the generator and the total e_s coefficient is nil for the turbine at normal n_s. In the case of a turbine with high n_s the total self-regulation coefficient may even be negative.

Without voltage control the grid load may vary considerably with the frequency-voltage ratio, and the total self-regulation coefficient may then vary between 2 and 5.

In a large grid, values are found between these limits according to the proportion of the different circuits and the ratio between frequency and voltage. Thus, according to the measurements reported by M. Nasse, a self-regulation value between 0·5 and 1·5 was obtained in the Paris grid, which corresponded to a total self-regulation coefficient of 1·5 to 2·5. For a limited network supplying an ohmic resisting circuit (e.g. electric furnaces) where the voltage is kept constant, the self-regulation coefficient may diminish to zero or may even become negative.

Fig. 2. **Stability conditions of general applicability.**



Improvement of stability limit by self-regulation.
With increasing self-regulation e_s the stability limit is reached with smaller rotating masses. The governing constants δ_t and δT_s, which allow the rotating masses (T_a) to be decreased, without self-regulation ($e_s = 0$) increase at the same time the effect of self-regulation to such an extent that with self-regulation parameters of 2 for dash-pot and 1 for acceleration-type governors, it would be possible, from the stability aspect, to reduce the rotating masses to any desired amount. (Full stability).

Effect of regulation constants on the damping period, without self-regulation ($e_s = 0$).
The smallest damping period $T_{1/10}$ is about ten times the acceleration period T_l of the water column. With increasing regulation constants δ_t and δT_s, the damping period $T_{1/10}$ decreases at first and then increases. The most favourable choice of regulation constants δ_t, T_i or δT_s, T_{ac} may also be determined from the curves.

e_s = Self-regulation constant ;
$T_{1/10}$ = Damping period to 1/10 of initial value ;
z_0 = Load factor related to power produced ;
z = Load factor related to water consumed ;
T_a = Acceleration period of rotating masses in sec. ;
T_l = Acceleration period of water column in sec. ;
T_s = Opening time of servo-motor running at maximum speed ;
δ = Speed deviation necessary to give maximum speed corresponding to T_s (related to average speed) ;
$\tau' = \delta T_s$ = Delay in sec. ;
T_i = Dash-pot rigidity = time to displace the dash-pot through its whole stroke, x_{max}, under the force (assumed constant) of its spring corresponding to deflection x_{max}.
δ_t = Temporary speed deviation necessary to change the servo-motor from idling to full load condition, dash-pot throttle assumed fully closed ($T_i = \infty$) ;
T_{ac} = Characteristic time of acceleration-type governor.

DAMPING PERIOD.

We have seen how self-regulation may assist automatic governing. On the other hand rapid damping causes difficulties in governing, the importance of which it will be attempted to assess. The damping period theoretically is infinitely long, the amplitudes diminishing asymptotically with time.

As a practical damping period the time required for all the governing components to be damped to at least 1/10, is introduced. This appears to be the best criterion for governing, in theory and in tests. In practice it is more important that damping should be quick enough to prevent another noticeable variation taking place in the meantime.

STABILITY CURVES TAKING ACCOUNT OF SELF-REGULATION AND DAMPING.

The curves shown in Fig. 2 are obtained from theoretical calculations. They show quantitatively, for all possible applications, the extent to which self-regulation will assist automatic governing (left) and to what extent this governing becomes difficult when it is attempted to reduce the damping period (right). They allow the determination of the governing constants necessary to achieve stability with a minimum WR^2 value. In spite of identical operation of the dash-pot and acceleration type governors, it is useful to express the results by means of the governing constants used

in practice for each of the two types. Thus the upper curves apply to the dash-pot and the lower ones to the acceleration-type. The simpler of the two identities implies that the time representing the rigidity of the dash-pot T_t should be as great as the time representing the characteristic acceleration time T_{ac}. This equation makes it possible to use the same abscissa in the two diagrams.

By eliminating dimensions we obtain curves of general validity. In the relative values usually employed the dimensions are eliminated, with the exception of time, and by introducing suitable time-ratios, this last dimension may also be dispensed with.

The result is that in the varied conditions presented by practical operation, the introduction of governing constants into these curves is sufficient to predict the operation of the governor. A comparison of the experimental results obtained in existing power stations with the results deduced from these curves will give a relatively narrow range of correction factors since these curves take into account the self-regulating effect and damping. Moreover, the relative values which only affect the maintenance of stability conditions can be ascertained directly.

With regard to the ordinates, the parameters chosen for the curves were : on the left-hand side of Fig. 2, the principal values of the self-regulating effect, and on the right, the damping period $T_{1/10}$ in relation to the hydraulic inertia.

It can be stated in accordance with the left hand curves that the term z is in the numerator of the abscissa as well as in that of the ordinate. For partial loads where $z < 1$, the governing coefficients, being in the denominator, may also be smaller. Thus, stability reaches a critical value at full load, but improves as the load is decreased. It is then permissible to reduce the value of the governing constants T_t, T_s, T_{ac} in the range of partial loads, if this procedure is convenient.

When the main valve is closed we use only the minimum quantity of water required for idling so that $z \ll 1$. This is why stability is easily obtained when the turbine is not connected to the grid. The self-regulation parameter indicates that stability will improve, first of all with increasing values of the self-regulation coefficient. It follows that in a circuit which is strongly self-regulating, it is possible to make do in spite of an unfavourable T_a/T_t ratio. This is due to the fact that during operation in parallel, the self-regulating effect assists the governor; the most favourable idling conditions are obtained before the connexion in parallel and after the circuit is disconnected. However, the conditions become difficult when a heavily loaded alternator has to supply a circuit with small self-regulation.

Referring to the curves the ordinates of which are proportional to the mechanical inertia T_a, it will be seen that the rotating masses WR^2 may be smaller as the self regulation parameter is increased.

Apart from the self-regulation constants e_s and the load factor z, the following values govern the effectiveness of self-regulation.

(a) the temporary speed deviation δ_t.
(b) the rigidity of the dash-pot T_t.
(c) the delay of the acceleration-type governor δT_s.

This explains why self-regulation is much more effective for hydraulic turbines than for steam turbines. In dash-pot governors there is a considerable temporary deviation whereas steam turbines usually have a short permanent deviation. The self-regulation effect, based on variation in speed, is then greater simply by reason of the larger deviation. Self-regulation becomes more effective as the control is effected more slowly. This affects, in dash-pot governors, the time representing the rigidity, and in acceleration-type governors, the delay which is over 1 second, instead of, for example, 1/20 of a second for steam turbines. Moreover, self-regulation is more effective in hydraulic turbines because its effect increases with the degree of loading just like the pressure impact. These facts show the importance of the self-regulating effect for hydraulic turbines, an effect which is negligible for steam turbines.

Given the mechanical inertia T_a we may see from the curves that, without self-regulation, the only means of improving stability would consist of increasing the values of the temporary deviation δ_t (above) and of the delay δT_s (below). This same means will also improve the effect of self-regulation.

This indicates that by ample self-regulation it is possible, at least under high load, to reach the stability limit without taking account of the rotating masses. On the limiting curves for the self-regulation parameter having a value of 2 for a dash-pot governor, and 1 for an acceleration-type governor, stability is maintained as far as the zero ordinate.

The mechanical inertia T_a may be eliminated, i.e. the rotating masses need not be taken into account, to ensure stability, when self-regulation and the load are large. At full load and with a self-regulation constant of $e_s = 4$, it will be seen that the rotating masses may be reduced to zero when the deviation is greater than 50 per cent in the case of the dash-pot governor ; and with an acceleration type governor and a hydraulic inertia of 3 seconds, for example, the rotating masses may be reduced to zero when the delay is more than 3/4 second.

Although such values are easily realized, it is not intended that the rotating masses should be reduced to arbitrary values in practice. A limit is rapidly reached where an inadmissible variation in the speed is produced. On the other hand, the conclusion that it is worth while producing a self-regulating effect in critical cases where it does not exist already, is justified.

The self-regulation coefficient is zero only when a resisting circuit is supplied and when there is instantaneous voltage control. If the exciting generator is coupled to the alternator, the voltage varies markedly with the speed. To achieve stability, it is sufficient, for small deviations from the stable condition, to use this natural relationship between the speed and voltage to create artificial self-regulation and when necessary this relationship may be amplified. Nevertheless, beyond the range of these very small variations in voltage, it is possible to maintain a constant voltage almost independently of the speed. Moreover, it is possible to arrange for a delay in the variation of the voltage by delaying the operation of the voltage control within the admissible range. All the alternators in the circuit must be controlled according to the same frequency-voltage law, because otherwise only the resisting torques between the machines vary with the voltage.

In larger circuits, the self-regulating effect is generally sufficient and usually only separate circuits fed by few groups of alternators are purely ohmic resisting. It is possible, however, with only a few groups, to make all the voltage controls operate approximately according to the same frequency-voltage law.

Even in the case of separate circuits, where an artificial self-regulation effect is introduced, the self-regulation coefficients will often remain smaller than in the case of much larger circuits.

The curves make it possible to ascertain, in advance, the smallest possible delay when the value for rotating masses is given. The different units can be expected to maintain the same stability whether they are connected to a grid having a self-regulation constant, of say 1·5, or to a small separate circuit where the self-regulation constant will reach only 0·5.

From the curves it can be deduced that the delay should be at least 4·8 seconds for the separate circuit whilst for that coupled to the more powerful self-regulating circuit a delay of 2 seconds will be sufficient for stability to be maintained. This applies to the

difficult case where the mechanical inertia is of the order of 3 seconds and the hydraulic inertia has the same value. As in a separate circuit the demand for constant frequency is usually not very exacting, a delay of 4·8 seconds is not inconvenient. On the other hand, when the frequency has to be maintained between fine limits in a large circuit, it is desirable to keep the delay to a minimum. In order to be able to reduce the rotating masses to the minimum it would be convenient to develop the governors in a way to make it possible to change the constants when changing over from operation in a separate circuit to operation in the grid. For separate operation it is then sufficient to have a small artificial amplification of the self-regulating effect.

Self-regulation is particularly effective in some critical cases, as for example when the mechanical inertia is reduced to 3 seconds and is not greater than the hydraulic inertia. It allows the problem to be solved in those critical cases when there is only a small self-regulation constant. This fortunate circumstance is explained by the fact that when the self-regulation is zero a large temporary deviation and a long delay is required to achieve satisfactory stability, with small rotating masses. As shown by the self-regulation parameters, however, a large temporary deviation δ_t or a long delay increases the self-regulation effect, although very small values of the self-regulation constant will produce considerable improvement.

Given the ever-increasing difficulties of the control problem arising from an increasing hydraulic inertia and a decreasing mechanical inertia, the tendency is to retard the action of the governor. This makes the self-regulation effect stronger. Conversely, when applied to airscrews, self-regulation will be decreasingly effective, because in this case there is the opposite tendency towards more and more rapid governing.

It will be seen from Fig. 2 (right) that only under conditions of undamped oscillations ($T_i/zT_l = \infty$) is it permissible to reduce continuously the time representing the rigidity T_i by a greater temporary deviation δ_t. From what has been said, the false conclusion must not be drawn that the action of the governor is more rapidly damped within very wide limits when the temporary deviation is increased. With a time representing the rigidity of, for example, $T_i/zT_l = 4$, the damping period falls rapidly at first but increases again with increasing deviation.

This has long been experienced and has led to some uncertainty as to the most favourable value of temporary deviation. This uncertainty is removed by data provided by the curves.

It will be noted that the damping period $T_{1/10}$ must always be an important multiple of the hydraulic inertia, in both methods of governing. The conditions of damping for the components of the governor operation limit the range of the damping period $T_{1/10}/zT_l$. It is thus impossible to reduce at will the damping period if the hydraulic inertia values are large. As there must always be a certain margin, a useful rule is that the shortest damping period is 10 times greater than the hydraulic inertia. Then, when there is a high hydraulic inertia value, say 3 seconds, the damping period must be at least 30 seconds.

The unfavourable values of the damping period due to an excessive delay and the impossibility of rapid damping are explained as follows :—

Governing is performed more slowly to damp out pressure impact. This damps one of the governor components more quickly but retards the damping of the other component so that it is damped less than the first, with the result that the damping period increases again.

For these reasons it is well worth while to co-ordinate the two values :—the temporary deviation and the rigidity of the dash-pot governor, or the delay and the characteristic time of the acceleration-type governor. The curves will assist in finding the most favourable co-ordination.

GUARANTEE FROM OVERSPEEDING.

The conditions under which it will be possible to make do with the inertia corresponding to the " natural WR^2 " value have been sought, by abandoning existing traditions. From this point of view we have to abandon the usual practice which, without being necessary for the maintenance of the stability and frequency, is opposed to reduction in the rotating masses. The users may allow the designers to increase the overspeeding guarantees to the extent permissible to maintain the frequency. It is necessary to find by experience the maximum delay with which the frequency can still be kept within permissible limits. Observations made from experience have shown that the maintenance of the frequency depends not only on the automatic governor but also on the manual control.

MANUAL CONTROL.

With the tendency to larger delays the present conception of governors is against effective and rapid manual overriding control by the staff. Whereas for connexion in parallel, for the maintenance of the frequency and for the distribution of the load, the staff should be able to set the governors as quickly and accurately as possible by means of a change in speed, this operation is made slower and its accuracy is diminished by the adoption of increasing delay values.

Even in the case when frequency is controlled by only some of the machines in the circuit, and a shorter permanent deviation is given to them than to the others, operation in parallel demands that this deviation should not be less than 2 per cent, i.e. than 1 cycle per second maximum deviation. These machines must then, in

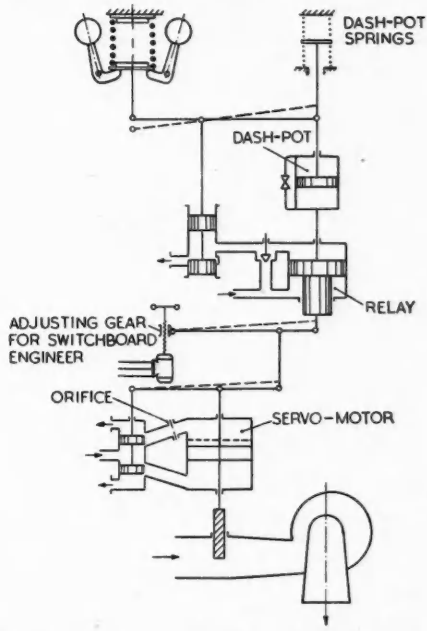


Fig. 3. **Speed governor with dash-pot and rapid manual overriding control.**

The main servo-motor can follow the governing movement as rapidly as desired, since the dash-pot provides the delay in regulation necessary for stability. The switchboard engineer can adjust the high speed servo-motor without delay due to the dash-pot.

order to adjust the whole circuit, supply power varying between almost zero and full load, which would correspond to frequency variations of one cycle without a correcting manual control, since an accuracy of 1/10 cycle is sought. In practice the tendency is to achieve this by manual operation of the change speed mechanism. The quality of the frequency regulation thus depends, to a great extent, on the rapidity and accuracy with which the automatically controlled machine will respond to the overriding manual control. The possibility of a new obstacle to the increase in the delay is thus apparent.

In modern governors, the servo-motor responds less quickly to adjustments made by the staff, when the delay is increased, because the change-speed motor acts first on the governor. Thus, the servo-motor will only respond to the adjustment with the retardation imposed by the degree of the delay adopted.

It is possible, however, while allowing the governor to operate with the delay demanded by stability, to make the change-speed motor actuate the servo-motor directly, i.e. without any intermediate retardation and without affecting the stability.

As explained for the dash-pot type governor, the servo-motor follows the movement of the distributor valve almost without delay. It is sufficient, therefore, to make the motor operate directly on this component.

Fig. 3 shows an example of application. The dash-pot of the tachometer group is not operated by the servo-motor but by a relay controlling the latter. The delay required for stability is achieved by the dash-pot operated by the relay. The servo-motor can thus follow the relay without any supplementary retardation. The remote-controlled motor which directly operates the distributor valve of the servo-motor thus allows instantaneous manual control which is in no way reduced by the delay of the dash-pot. Limitation of the maximum speed by means of an orifice which limits pressure impact does not prevent rapid overriding manual control since this consists of small load adjustments. Change in load thus follows the manual control with a negligible delay.

It is, therefore, possible to increase the delay without preventing rapid overriding manual control.

CONCLUSION.

To summarize the conclusion drawn from this investigation of the effects of a reduction in the rotating masses of hydro-electric power plant :—

(1) It can be calculated quantitatively from the non-dimensional curves of general applicability the extent to which self-regulation will improve automatic governing, and the extent to which the demand for more rapid damping will make the governing process more difficult to realize.

Non-dimensional values show directly the ratios which should exist to make possible the adaptation of governors to the very varied requirements of practical operation.

(2) The self-regulation of hydraulic turbines by speed variation is much more effective than for steam turbines by reason of the fact that with the dash-pot governor the temporary deviation is much greater, while with the acceleration- type governor the speed of regulation is much less. Self-regulation also increases in proportion with the load factor as well with the pressure impact the effects of which it must resist.

The relative effect of self-regulation thus remains the same, whatever the load. Calculation indicates that it would be possible to achieve stable governing of a group of machines having zero mechanical inertia if the degree of self-regulation and loading were sufficiently high.

3) The damping period cannot be reduced arbitrarily. It will be at least 10 times greater than the time representing the hydraulic inertia.

Pronounced self-regulation facilitates governing to a much greater extent than a damping period, reduced to the absolute minimum, makes it difficult.

Governing cannot be improved indefinitely by progressively increasing the temporary deviation of dash-pot governors or the closure time of acceleration-type governors. The damping period decreases first and then increases again.

Between the characteristic values of the same governor (δ_t, T_i/T_a, T_{ac}) there are optimum ratios which may be read directly from the curves.

(4) For equal delay values, the dash-pot and acceleration-type governors are equivalent, from the point of view of stability and damping.

By increasing the delay, even when the hydraulic inertia is large, it is possible to reduce the mechanical inertia to the minimum so long as this delay is not excessive for the proper maintenance of the frequency.

(5) With modern governors, large delay values prevent the maintenance of the frequency because they reduce the possibility of overriding manual control. This obstacle to the increase of the delay value may be eliminated by making the remote-controlled motor act directly on the distributor valve of a rapid servo-motor.

(6) Reduction of mechanical inertia implies the adoption of wider guarantees for variations in speed when the load is decreased.

(7) Large delay values which enable small mechanical inertia values to be successfully used, make the maintenance of the frequency difficult. In a large grid there will generally be sufficient self-regulation to make it possible, with moderate delay values, to obtain both stability and maintenance of the frequency, with the mechanical inertia required, by design alone.

In a separate circuit of purely ohmic resisting characteristics, large governing delays are required if the voltage is to be maintained at a constant value.

On the other hand, the requirements relating to the maintenance of the frequency are generally less exacting in separate circuits. As there are then few groups of generators, it should be possible, by controlling the voltage according to a frequency-voltage law common to all groups, to amplify the self-regulation effect without disturbing the voltage from its prescribed value beyond an upper and a lower limit.

In the case of alternator groups operating both as separate circuits and connected to the grid, it may be advantageous to switch over to other values for regulator constants.

It is precisely in the most critical cases, where the hydraulic inertia is great and the mechanical inertia small, that the smallest self-regulation coefficients give a considerable improvement.

Tests involving all the methods described should show, for various types of loading and for the varied conditions of the circuit, to what point stability of governing is compatible with satisfactory maintenance of the frequency without the necessity to supplement the minimum mechanical inertia, required by the design, by the effect of additional weight.

This goal is not yet reached and for the present a reduction in the mechanical inertia to the optimum intermediate value must suffice. However, so long as users and manufacturers and electrical and hydraulic engineers work together, it can be expected that the conditions of governing which still oppose a more economical development of hydro-electric power plant will be progressively eliminated.

Vibrations of Steam Turbine Blades

By A. V. LEVIN. (From a collection of Papers on the *Dynamic Strength of Machine Parts*, Academy of Sciences, U.S.S.R., Institute of Mechanical Engineering, Moscow, 1946, pp. 25-43, 7 illustrations.)

THE disks and blades of a turbine rotor such as shown in Fig. 1, operating at speeds up to 3,000 r.p.m. are considerably stressed by centrifugal forces, especially in the last stages, and the blades are subjected to static bending by the steam forces. The blades vibrate about the minor axis of inertia of the section almost in the plane of the wheel (i.e. tangentially). As the disks are stiffest in this direction, these vibrations are not shared by them to a very appreciable degree and are confined to the blade packets only.

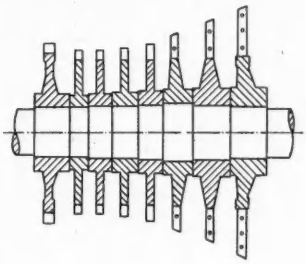


Fig. 1

Blade oscillations about the major axis of inertia of the section are approximately at right angles to the plane of the wheel and therefore are simultaneous with those of the disks.

The dangerous axial vibrations of a disk with its blades are in the nature of nodal diameters. The frequencies of the exciting forces are known, but their magnitude can vary within fairly wide limits, depending on the finish of the diaphragms.

The blades of the last stages, which have comparatively low frequencies, are tuned, i.e. the frequencies of the natural vibrations of the blade packet are chosen so as to leave a sufficiently wide margin from resonance. If this is not possible, the stresses in the working blades are determined by considerations of the possibility of their working in resonance conditions. Owing to the long service life of stationary turbines (i.e. up to 100,000 hours), possible changes in the strength characteristics of the blades must be considered.

Experimental investigations in progress in the Stalin Works have shown that the fatigue limit of blades falls sharply in operation, and that nickel steel blades could only sustain 10 million cycles after 94,000 hours in service under a stress of 14 kg/mm² or less, whereas when new they had a fatigue limit of 35 kg/mm². Blades of untempered steel decreased on an average to half the original value after 12,000 hours in service.

As the magnitudes of the exciting forces producing blade vibrations are unknown, the dynamic stress calculation reduces to a consideration of relative stresses in the elements of a packet which allows a comparative estimate of the strength of the individual components.

Table I gives the calculated results for a packet of blades in static and dynamic bending, and the symbols are explained in Fig. 2, viz. :

m is the number of blades in the packet ;

M_m' and M_m'' are the moments acting in the shrouding applied to the m-th blade of a packet from the left and right, respectively ;

M_m is the moment exerted by the shrouding on the m-th blade ;

θ_m is the shear force acting on the m-th blade at the point of attachment to the shrouding ;

$\overline{M}_m$ is the moment acting at the root section of the blade.

In column 2 the relative values of M_m are given on the assumption of a unit moment acting on the cross-section of the centre blade. In columns 3 and 4 the relative values of moments M_m (again taking those for the centre blade as a unit) are given together with those at the root.

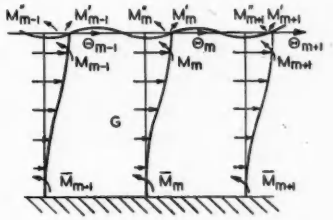


Fig. 2

The table shows that the distribution of moments in the static case and during oscillations is practically identical. Yet as a criterion for the suitability of the packet of blades the values in columns 2 and 4 are insufficient. It is not enough to have a picture of the distribution of moments along the shrouding and the root sections of the blades and it is necessary to know their absolute values, or the ratios of stresses in the shrouding and root sections of the corresponding blades. In the columns under 5 these ratios are given and they show that the results for the static condition and during vibration no longer coincide. The difference in the values of the two ratios is up to 40 per cent, and it is therefore impossible to judge the behaviour of vibrating blades from the figures for their static condition. The following gives the method of calculation practised in the Stalin Works.

TABLE I

1	2		3		4		5	
m	M_m'		M_m		$\overline{M}_m$		σ_b/σ_A	
	stat.	dynam.	stat.	dynam.	stat.	dynam.	stat.	dynam.
1	1·692	1·700	0·846	0·850	0·965	0·941	0·417	0·590
2	0·850	1·845	1·035	1·031	1·009	1·010	0·200	0·273
3	1·035	1·035	0·993	0·992	0·999	0·997	0·246	0·339
4	0·993	0·994	1·002	1·001	1·000	0·998	0·236	0·324
5	1·004	1·002	1·000	0·999	1·000	1·000	0·238	0·327
6—15	1·000	1·000	1·000	1·000	1·000	1·000	0·238	0·326
16	0·999	0·998	1·000	0·999	1·000	1·000	0·237	0·326
17	1·010	1·010	1·000	1·001	1·000	0·998	0·239	0·330
18	0·954	0·952	0·993	0·992	0·999	0·997	0·227	0·311
19	1·220	1·220	1·035	1·031	1·009	1·010	0·288	0·334
20	0·000	0·000	0·846	0·850	0·965	0·941	0·000	0·000

1. THE BLADES.

Calculation of the Stationary Blades.

As the calculations for dynamic strength are based on those for static bending, we start with a survey of the latter.

The calculations for static bending and vibration of the blades may be carried out as for a finite number of blades united in a packet on the assumption that they all work under similar conditions. In this way sufficiently accurate figures for the frequencies of oscillations of the packet, and the mean values of the stresses associated with them, are obtained.

Considering the static bending of a blade packet, held together by shrouding and two lacing wires, it is possible to refer to one blade separated from the others which in bending is acted upon by uniform steam pressure and the bending moments imparted by the lacing wires and shrouding (see Fig. 3). On the assumption that all the blades work under identical conditions, the moments are :

$$M_i = \frac{12\,EH_i\,I_i}{t_i}\,y'(l_i) = \frac{k_i\,EI_0}{l}\,\eta_i$$

$$k_i = \frac{12\,H_i\,I_i\,l}{t_i\,I_0}\;;\quad \eta_i = y'(l_i)$$

where

- I_i is the moment of inertia of the section of the lacing wire (shrouding) ;
- t_i is the pitch along the lacing wire (shrouding) ;
- I_0 is the moment of inertia of the root section of the blade ;
- l is the length of the blade not including the root ;
- $y'(l_i)$ is the slope of the deflected axis of the blade in its point of attachment to the wire (shrouding) ;
- H_i is a coefficient allowing for the rigidity of the attachment of the wire to the blade, determined empirically.

The bending equations of the blade may be written :

$$0 \leqslant x \leqslant x$$

$$y'' = \frac{M_0}{EI_0}f''(x) - \frac{k_1}{l}\eta_1\phi''(x) - \frac{k_2}{l}\eta_2\phi''(x) - \frac{k_b}{l}\eta_b\phi''(x)$$

$$x_1 \leqslant x \leqslant x_2$$

$$y'' = \frac{M_0}{EI_0}f''(x) - \frac{k_2}{l}\eta_2\phi''(x) - \frac{k_b}{l}\eta_b\phi''(x)$$

$$x_2 \leqslant x \leqslant 1$$

$$y'' = \frac{M_0}{EI_0}f''(x) - \frac{k_b}{l}\eta_b\phi''(x)$$

$$M_0 = \frac{ql^2}{2},\quad f''(x) = (1-x)^2\frac{I_0}{I},\quad \phi''(x) = \frac{I_0}{I}$$

Integration of these equations yields :—

$$0 \leqslant x \leqslant x_1$$

$$y' = \frac{M_0\,l}{EI_0}f'(x) - k_1\eta_1\phi_1'(x) - k_2\eta_2\phi_2'(x) - k_b\eta_b\phi_b'(x) + C_1$$

$$x_1 \leqslant x \leqslant x_2$$

$$y' = \frac{M_0\,l}{EI_0}f'(x) - k_2\eta_2\phi_2'(x) - k_b\eta_b\phi_b'(x) + C_1$$

$$x_2 \leqslant x \leqslant 1$$

$$y' = \frac{M_0\,l}{EI_0}f'(x) - k_b\eta_b\phi_b'(x) + C_1$$

$$f'(x) = \int_1^x f''(x)\,dx\;;\quad \phi_1'(x) = \int_{x_1}^x \phi''(x)\,dx\;;$$

$$\phi_2'(x) = \int_{x_2}^x \phi''(x)\,dx\;;\quad \phi_b'(x) = \int_1^x \phi''(x)\,dx$$

and $f'(x)\ldots\phi_b'(x)$ may easily be evaluated by the trapezium rule.

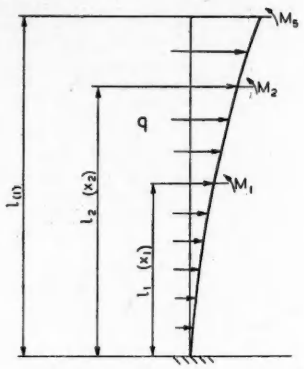


Fig. 3

The arbitrary constant C_1 and the slopes of the blade axis at the points of attachment of the wires are found from the equations

$$x = 0;\quad y' = 0\;;$$
$$x = x_1;\quad y' = \eta_1;$$
$$x = x_2;\quad y' = \eta_2;$$
$$x = 1:\quad y' = \eta_b.$$

Integrating further, an expression for the deflection curve is obtained

$$0 \leqslant x \leqslant x_1$$

$$y = \frac{M_0\,l^2}{EI_0}f(x) - k_1\eta_1 l\phi_1(x) - k_2\eta_2 l\phi_2(x) - k_b\eta_b l\phi_b(x) + C_1 x l + C_2$$

$$x_1 \leqslant x \leqslant x_2$$

$$y = \frac{M_0\,l}{E\,l_0}f(x) - k_2\eta_2 l\phi_2(x) - k_b\eta_b l\phi_b(x) + C_1 x l + C_2$$

$$x_2 \leqslant x \leqslant 1$$

$$y = \frac{M_0\,l}{E\,I_0}f(x) - k_b\eta_b\phi_b(x) + C_1 x l + C_2$$

$$f(x) = \int_1^x f'(x)\,dx\;;\quad \phi_1(x) = \int_{x_1}^x \phi_1'(x)\,dx\;;$$

$$\phi_2(x) = \int_{x_2}^x \phi_2'(x)\,dx\;;\quad \phi_b(x) = \int_1^x \phi_b'(x)\,dx$$

C_2 is determined by

$$x = 0,\ y = 0.$$

With η_1, η_2 and η_b known, the stresses in the sections of the blade, the lacing wires, and the shrouding, can be found :—

$$\sigma_\pi(x) = \frac{EIy''(x)}{W}$$

where I is the moment of inertia of the blade section, and W is its section modulus.

The stresses in the wire sections (shrouding) are found from

$$\sigma_i = \frac{M_i}{2\,W_i}$$

where W_i is the section modulus of the wire section (shrouding).

Having ascertained the curve of static deflection of the blade, the frequency of the oscillations of the packet may be calculated by Rayleigh's method, thus

$$p^2 = \frac{2\,EI_0\,g}{\gamma\,l^4} \times \frac{\int_0^1 \frac{z(x)}{z(1)}\,dx}{\left\{\int_0^1 F(x)\left[\frac{z(x)}{z(1)}\right]^2 dx + \sum \frac{V_i}{l}\left[\frac{z(x)_i}{z(1)}\right]^2\right\} z(1)}$$

where $$z(x) = y(x) \Big/ \frac{M_0\,l^2}{EI_0}$$

V_i being the circular pitch of the wire, and $F(x)$ the equation of the boundary curve of the cross-section of the blade.

Repeated computations have shown that the frequencies obtained by this formula do not differ by more than two per cent from the results obtained by numerical integration of the differential equation of the oscillations.

The method of successive approximations is used for the calculation of the dynamic stresses (i.e. stress ratios). Let us consider the bending of the packet under a distributed load, namely the inertia forces. A first approximation to the curve of deflection by oscillations is made by taking a curve the ordinates of which are proportional to the deflections in static bending under uniform pressure :

$$Y(x) = C\frac{y(x)}{y(1)}$$

The moment acting in the cross-section at a distance x of the root end (Fig. 4) under the distributed load, and to the right of it, is

$$M = \frac{\gamma}{g}\,p^2\,l^2\,C\int_x^1 F(\rho)\,\frac{y(\rho)}{y(1)}\,(\rho - x)\,d\rho$$

The moment exerted by the wire on the cross-section x is

$$\overline{M}_i = \frac{\gamma}{g}\,p^2\,l^2\,C\,\frac{V_i}{l}\,\frac{y(x_i)}{y(1)}\,(x_1 - x) - k_i\,\frac{EI_0}{l}\,X'(x_i);$$

$$k_i = \frac{12\,H_i\,I_i\,l}{t_i\,I_0}$$

where x is the coordinate of the location of the wire, V is the circular pitch of the wire, $X'(x_i)$ is the deflection of the axis of the blade during the oscillation at the point of attachment of the wire.

We introduce the following symbols :—

$$\frac{\gamma}{g}\,\rho^2\,l^2\,C = M_0'; \quad \frac{I_0}{I}\int_x^1 F(\rho)\,\frac{y(\rho)}{y(1)}\,(\rho - x)\,d\rho = \Phi''(x);$$

$$\frac{I_0}{I} = \phi''(x); \quad (x_i - x)\,\frac{I_0}{I} = \psi_i''(x); \quad X'(x_i) = \Theta_i$$

The differential equations of the bending of a blade of a packet fastened, say, by two lacing wires and a shrouding, may be written

$$0 \leqslant x \leqslant x_1$$

$$X'' = \frac{M_0'}{EI_0}\left\{\Phi''(x) + \frac{V_1}{l}\,\frac{y(x_1)}{y(1)}\,\psi_1''(x) + \frac{V_2}{l}\,\frac{y(x_2)}{y(1)}\,\psi''(x) + \frac{V_b}{l}\,\psi_b''(x)\right\} - \frac{k_1}{l}\,\Theta_1\,\phi''(x) - \frac{k_2}{l}\,\Theta_2\,\phi''(x) - \frac{k_b}{l}\,\Theta_b\,\phi''(x)$$

$$x_1 \leqslant x \leqslant x_2$$

$$X'' = \frac{M_0'}{EI_0}\left\{\Phi''(x) + \frac{V_2'}{l}\,\frac{y(x_2)}{y(1)}\,\psi_2''(x) + \frac{V_b}{l}\,\psi_b''(x)\right\} - \frac{k_2}{l}\,\Theta_2\,\phi''(x) - \frac{k_b}{l}\,\Theta_b\,\phi''(x) \quad \ldots (1)$$

$$x_2 \leqslant x \leqslant 1$$

$$X'' = \frac{M_0'}{EI_0}\left\{\Phi''(x) + \frac{V_b}{l}\,\psi_b''(x)\right\} - \frac{k_b}{l}\,\Theta_b\,\phi''(x)$$

Integrating twice and allowing for the following conditions

$$x = 0\begin{cases} X = 0 \\ X' = 0 \end{cases}$$
$$x = x_1,\ X' = \Theta_1$$
$$x = x_2,\ X' = \Theta_2$$
$$x = 1,\ X' = \Theta_b$$

the deflected curve of the blade and the numerical values of Θ_i are found. Therefore, all the data necessary for calculating the stresses (i.e. stress ratios) in the individual elements of the packet are available.

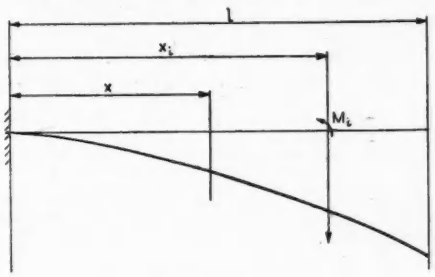


Fig. 4

When necessary, the following approximation may be used for solving the problem of the deflection of blades under inertia forces, wherein the first approximation may serve as the initial deflection curve.

As the calculations show, the stress ratios obtained by the first approximation differ by only a few per cent from those obtainable by numerical integration of the differential equation of the oscillations, so that there is no need for a further approximation.

When computing the blade packets by the method of successive approximations for the first harmonic of the tangential oscillations, the initial curve to be used is one of the following form :—

$$X_{20} = X_0 - A_1 X_1$$
$$X_0 = 3x^3 - 2x^2$$
$$x = 0,\ X_0 = 0,\ X_0' = 0$$

where X_1 denotes the deflection curve of the fundamental of the oscillation of the packet, as obtained by the first of the successive approximations, and A_1 is a constant factor determined from the condition

$$\int_0^1 FX_{20}\,X_1\,dx = -\sum \frac{V_i}{l} X_{20}(x_i)\,X_1(x_i)$$

where the summation sign is to be extended over all the wires (shrouding) of the packet.

Then

$$A_1 = \frac{\int_0^1 FX_0\,X_1\,dx + \sum \frac{V_i}{l} X_0(x_i)\,X_1(x_i)}{\int_0^1 FX_1^2\,dx + \sum \frac{V_i}{l} X_1^2(x_i)} \quad .. \quad (2)$$

Having determined A_1 and, therefore, knowing X_{20}, the problem of the deflection of the blade packet under the action of inertia forces is solved by using the curve X_{20} as the deflection curve. The system of differential equations will be analogous to the system (1), with the only difference that $y(x)$ is replaced by X_{20}.

By solving this system the deflection curve X_{21} is found, and to improve it we form the curve

$$y_{21} = X_{21} - A_2 X_1$$

A_2 can be determined from the condition analogous to eq. (2),

$$A_2 = \frac{\int_0^1 FX_{21}\,X_1\,dx + \sum \frac{V_i}{l} X_{21}(x_i)\,X_1(x_i)}{\int_0^1 FX_1^2\,dx + \sum \frac{V_i}{l} X_1^2(x_i)}$$

With y_{21}, the stresses (i.e. stress ratios) and the frequency of vibrations of the packet can be calculated by using the formula

$$p_2 = \frac{Eg}{\gamma l^4} \frac{\int_0^1 I(y_{21}'')^2\,dx + \sum k_i\,I_0\,(y_{21}')^2_{x=xi}}{\int_0^1 F(y_{21})^2\,dx + \sum \frac{V_i}{l}\,[y_{21}(x_i)]^2}$$

where $$k_i = \frac{12\,H_i\,I_i\,l}{t_i\,I_0}$$

As shown by the actual calculations and experimental data curve y_{21} is sufficiently accurate, and no further approximation is necessary.

CALCULATION OF THE ROTATING BLADES.

The effect of the centrifugal forces acting on the blades is to increase the vibration frequencies as a result of the stiffening of the blades.

The differential equation of the blade oscillations takes the form

$$E(IX'')'' = \frac{\gamma}{g} l^4 p^2 FX + \frac{\gamma}{g} \omega^2 l^4 \left\{ FX - \right.$$
$$- F\left(\frac{R_0}{l} + x\right) X' + \left[\int_x^1 F(\rho) \left(\frac{R_0}{l} + \rho\right) d\rho + \right.$$
$$\left. \left. + \frac{V_0}{l} \left(\frac{R_0}{l} + 1\right) X'' \right] \right\} \quad .. \quad .. \quad .. \quad (3)$$

The boundary conditions for a shrouded packet of blades may be written

$$x = 0, X = 0, X' = 0$$

$$x = 1, \quad IX'' = -\frac{12 H_b\,I_b\,l}{t_b} X'$$

$$x = 1, \quad E(IX'')' = \frac{y}{g} V_b\,l^3 \left\{ \omega^2 \left[\left(\frac{R_0}{l} + 1 \right) X' - X \right] - p^2 X \right\}$$

where ω is the angular velocity of the rotor. Differential equation (3) can be solved by the method of successive approximations.

An expression for the frequency may be obtained either from the differential equation, or from the law of conservation of energy.

$$p^2 = \frac{EI_0 g}{\gamma l^4} \frac{\int_0^1 \frac{I}{I_0} (X'')^2\,dx + \sum k_i \left\{ X'(x_i) \right\}^2}{\int_0^1 FX^2\,dx + \sum \frac{V_i}{l} X^2(x_i)} + B\omega^2$$

Then:—

$$B = \frac{\int_0^1 F\left(\frac{R_0}{l} + x\right) dx \int_0^x (X')^2\,dx + \sum \frac{V_i}{l} \left(\frac{R_0}{l} + x_i\right) \int_0^{x_i} (x')^2\,dx}{\int_0^1 FX^2\,dx + \sum \frac{V_i}{l} X^2(x_i)} - 1$$

where the sums have to be extended over all the lacing wires of the packet.

As shown by the corresponding operations when evaluating p^2 and B, it is sufficient to use the curve of static deflection of the blade (as representative of all the blades of the packet).

2. DISKS.

The differential equation of the forced vibrations of the disk takes the form

$$\Delta D\,\Delta W - \frac{1-\mu}{r} \left\{ \frac{dD}{dr} \frac{\partial^2 W}{\partial r^2} + \frac{d^2 D}{dr^2} \left(\frac{\partial W}{\partial r} + \right.\right.$$
$$\left.\left. + \frac{1}{r} \frac{\partial^2 W}{\partial \phi^2} \right) \right\} = f(r\phi t) - \frac{2\gamma}{g} h \frac{\partial^2 W}{\partial t^2} \quad .. \quad (4)$$

$$\Delta = \frac{\partial^2}{\partial r^2} + \frac{1}{r} \frac{\partial}{\partial r} + \frac{1}{r^2} \frac{\partial^2}{\partial \phi^2};$$

$$D = \frac{2Eh^3}{3(1-\mu^2)}$$

where $2h$ is the variable thickness of the disk, μ is Poisson's ratio, W is the deflection of the mid-plane of the disk, and r is the disk radius.

Eq. (4) is obtained on the usual assumptions of elastic theory that the mid-plane of the disk is unstressed, and that a perpendicular to this plane at the disk centre after deflection remains perpendicular to the surface of bending.

For the inner contour of the disk the conditions of attachment to the shaft, and on the outer contour the influence of the blades have to be considered.

It is impossible to solve differential equation (4) for disks of varying thickness and the dynamic strength calculations as well as the determination of frequencies, are possible only by approximations.

Disk profiles which are of practical importance may be determined by the method of successive approximations.

The differential equation (4) for the static deflection of a disk of variable thickness under a load $f(r\phi)$ is of the form

$$\Delta D\,\Delta W-(1-\mu)\left\{\frac{d^2D}{dr^2}\left(\frac{1}{r^2}\frac{\partial^2W}{\partial\phi^2}+\frac{1}{r}\frac{\partial W}{\partial r}\right)+\right.$$
$$\left.+\frac{1}{r}\frac{dD}{dr}\frac{\partial^2W}{\partial r^2}\right\}=f(r\phi)$$

When the load is expressed by

$$f(r\phi)=q(r)\cos n\phi$$

and $W(r\phi)=X(r)\cos n\phi$,

$$\Delta D\,\Delta X-(1-\mu)\left\{\frac{d^2D}{dr^2}\left(\frac{1}{r}\frac{dX}{dr}-n^2\frac{d^2X}{dr^2}\right)+\right.$$
$$\left.+\frac{1}{r}\frac{dD}{dr}\frac{d^2X}{dr^2}\right\}=q(r)$$

where $$\Delta=\frac{d^2}{dr^2}+\frac{1}{r}\frac{d}{dr}-\frac{n^2}{r^2}$$

Assuming a hyperbolic profile (in the special case of constant cross-section)

$$h=\alpha/r^m$$

Transforming the variables in the original equation by the substitution

$$r=e^{\rho},\ X=U(\rho)\,e^{\frac{2+3m}{2}\rho}$$

gives

$$U^{IV}-2\alpha U''+\beta U=\frac{q(\rho)}{D_0}\,e^{\frac{3m+6}{2}\rho}$$

$$D_0=\frac{2E\alpha^3}{3(1-\mu^2)}\quad\ldots\quad\ldots\quad(5)$$

$$\alpha=1+n^2+1{\cdot}5(1+\mu)m+2{\cdot}25\,m^2,$$

$$\beta=\frac{(2+3m)^4}{16}+\frac{(2+3m)^2}{4}[2n^2-3m(1-\mu)+n^2]+$$
$$+n^2[n^2-2-3m-(1+3m)(2+3m\mu)]$$

The solution of eq. (5) presents no particular difficulties for the given loading $q(p)$ and disk profile, and may be written :—

$$U=\sum_{i=1}^{4}C_i\,U_i(\rho-\rho_i)+\int_{\rho_i}^{\rho}U_4(\rho-\eta)\,\phi(\eta)\,d\eta$$

where $\rho_i=\log_e r_i$, and r_i = the internal disk radius

$$\phi(\eta)=\frac{q(\eta)}{D_0}\,e^{(s+1{\cdot}5m)\eta}$$

	Φ	Φ'	Φ''	Φ'''
U_1	1	0	0	0
U_2	0	1	0	0
U_3	0	0	1	0
U_4	0	0	0	1

$U_1 \ldots U_4$ are four particular solutions of eq. (5) without disponible term, which for $\rho=\rho_i$ satisfy the unit matrix.

They are easily obtainable by Cauchy's general method

$$U_1=\frac{1}{2\sqrt{\alpha^2-\beta}}\left\{\lambda^2\cosh\sigma(\rho-\rho_i)-\sigma^2\cosh\lambda(\rho-\rho_i)\right\}$$

$$U_2=\frac{1}{2\sqrt{\alpha^2-\beta}}\left\{\frac{\lambda^2}{\sigma}\sinh\sigma(\rho-\rho_i)-\frac{\sigma^2}{\lambda}\sinh\lambda(\rho-\rho_i)\right\}$$

$$U_3=\frac{1}{2\sqrt{\alpha^2-\beta}}\left\{-\cosh\sigma(\rho-\rho_i)+\cosh\lambda(\rho-\rho_i)\right\}$$

$$U_4=\frac{1}{2\sqrt{\alpha^2-\beta}}\left\{-\frac{1}{\sigma}\sinh\sigma(\rho-\rho_i)+\frac{1}{\lambda}\sinh\lambda(\rho-\rho_i)\right\}$$

$$\lambda^2=\alpha+\sqrt{\alpha^2-\beta},\quad\sigma^2=\alpha-\sqrt{\alpha^2-\beta}$$

The arbitrary constants C_i are determined from the boundary conditions on either side of the profile of the disk. $M=0$ represents the special case of a disk of constant profile, and the problem is much simplified, because tables for the integrals U_i can be readily compiled. In the dynamic strength calculations, $q(r)$=const. is taken as a first approximation. This reduces the problem to the case of bending of the disk under a loading, constant over the radius and varying along the arc as $\cos n\phi$ or $\sin n\phi$.

The curve of bending thus obtained may be used for calculating the frequency of the disk vibrations (disk complete with its blades).

The calculation of the occurring stresses may, again, be based on the method of successive approximations and, taking the curve of static bending as the initial curve, the static deflection of the disk under a load proportional to the deflection due to static bending can be found. In this case, evidently, $q(\eta)$ will be proportional to

$$h(\eta)\,X(\eta)$$

where $2h(\eta)$ is the disk thickness, and $X(\eta)$ is the deflection under a static load constant over the disk radius.

Stodola's method of calculating the frequencies of the disks complete with blades is used only when the blades are very short, so that their own deflection is negligible in comparison with that tangential to the curve of bending of the disk over the external radius (at the point of attachment of the blade).

The calculation of disks of any given profile, with blades of any length, may be carried out according to the method given below.

Assuming, with Stodola, the curve of bending of the disk over the radius to be given by

$$f(r)=\alpha r^s$$

where α and s are constant parameters, determined as shown later, and r is the radius.

Assume a curve proportional to the bending curve in an axial direction under a uniform stress as the curve of natural deflection of the blades (radial), *viz.*

$$\beta\frac{y(x)}{y(1)}$$

where β is a constant, $y(x)$ is the static bending curve of the blade in an axial direction under uniform stress, $y(1)$ is the deflection of the blade top, and $y(x)$ is determined by the equation

$$EIy''=\frac{ql^2}{2}(1-x)^2\quad\ldots\quad\ldots\quad(6)$$

where I is the function of the maxima of moments of inertia of the blade section, l is the length of the blade, and q is the specific loading.

Integrating eq. (6) twice and satisfying the boundary conditions

$$x=0,y=0,y'=0$$

the required bending curve $y(x)$ is obtained.

Hence,
$$\frac{y(x)}{y(1)}=\frac{\Phi(x)-x\Phi'(0)-\Phi(0)}{-\Phi'(0)-\Phi(0)}$$

where
$$\Phi''(x)=(1-x)^2\frac{I_0}{I}$$

I_0 is the maximum moment of inertia of the root section of the blade.

$$\Phi'(x) = \int_1^x \Phi''(x)\,dx, \quad \Phi(x) = \int_1^x \Phi'(x)\,dx$$

The full deflection of the blade (in radial direction) is found by adding the contributions corresponding to the deflections tangential to the bending curve of the disk ($r = r_a$) and the static bending:

$$v = f(r_a) + (b + l\,x)\,f'(r_a) + \beta\frac{y(x)}{y(1)} =$$

$$= \alpha\left[1 + \frac{b + lx}{r_a}\,s\right] r_a^s + \beta\frac{y(x)}{y(1)}$$

$$0 \leqslant x \leqslant 1$$

THE DISK ROOT.

As shown by calculations and by empirical data, the disk root does not influence the frequencies of the disk vibrations where the number of nodal diameters is four and more. It has, however, some influence on the oscillation with three nodal diameters, and a very marked one on vibrations with two nodal diameters. In the latter case there is also an appreciable effect on the frequency of the elements fastening the disk to the shaft. A comparison of the calculated and experimental data shows that for two nodal diameters the computed values of the frequency are satisfactory when compared with the results of bench tests on disks, when it is assumed that the deformation of the disk root is insignificant, and it is sufficient to consider only the deflection along the periphery. The deflection of the disk root is assumed to be in a radial direction following the tangent to the bending curve of the disk, at $r = r_i$ (see Fig. 5).

For the moment acting in any cross-section

$$M = -\frac{B}{(r_i - b_2)^2}\,\frac{d^2 v_1}{d\phi^2}$$

where

$$B = EI_{BT}$$

I_{BT} is the moment of inertia of the disk root referred to the axis of the disk section, v_1 is the displacement of the C.G. of the disk root, and ϕ is the angle reckoned from any diameter.

From Fig. 6,

$$v_1 = [f(r_i) - b_2 f'(r_i)]\cos n\phi$$

where $f(r_i)\cos n\phi$ is the deflection, and $f'(r_i)\cos n\phi$ is the slope of the deflection curve of the section of the disk adjoining the root.

The potential energy is equal to

$$\Phi_\pi' = \frac{1}{2\,E\,I_{BT}}\int_0^{2\pi} M^2 (r_i - b_2)\,d\phi$$

Introducing for M and $f(r)$ their correct expressions, and integrating,

$$\Phi_\pi' = \frac{\alpha^2\,E\,I_{BT}\,n^4}{2\,(r_i - b_2)^3}\left(1 - \frac{b_2}{r_i}\,s\right)^2 \rho^{2s}\,\pi\,r_a^{2s}$$

The kinetic energy of the disk root is then

$$\Phi_k' = \frac{1}{2}\int_0^{2\pi} c^2\,dm$$

c is the peak velocity of the vibration of an element of the disk root

$$c = pv_1$$

p is the circular frequency of the bladed disk, and dm, is the mass of the root element

$$dm = \frac{\gamma}{g}F_{BT}(r_i - b_2)\,d\phi$$

F_{BT} is the cross-sectional area of the disk root.

Substituting in the integral the proper expressions for c, v_1, and dm, and integrating,

$$\Phi_k' = \frac{\gamma}{g}\,p^2\,\alpha^2\,\frac{F_{BT}(r_i - b_2)}{2}\left(1 - \frac{b_2}{r_i}\,s\right)\rho^{2s}\,\pi\,r_a^{2s}$$

THE DISK BODY.

The expressions for the potential and kinetic energy of the disk body may be written, as for a plate of variable thickness (cf. A. Stodola, Dampf-und Gasturbinen, 6th ed., §187, and S. P. Timoshenko, Theory of Oscillations, §56).

Allowing for the deflection of the disk

$$W = ar^s\cos n\phi :-$$

$$\Phi_\pi'' = \frac{E}{1-\mu^2}\,\frac{\pi\alpha^2}{3}\,[(s^2 - n^2)^2 -$$

$$-\,2\,(1-\mu)\,(s-1)\,(s^2 - 2\,n^2\,s + n^2)]\int_{r_i}^{r_a} h^3\,r^{2s-3}\,dr$$

$$\Phi_E'' = \alpha^2\,\frac{\pi\gamma}{g}\,p^2\int_{r_i}^{r_a} hr^{2s+1}\,dr$$

where $2h$ is the variable disk thickness.

The integrals on the right-hand side can easily be evaluated, using approximate formulae, for every case of interest, and for some particular disk profiles even analytically.

THE DISK RIM.

Considering the rim as a curved beam, or cantilever, the expression for the moment acting on its cross-

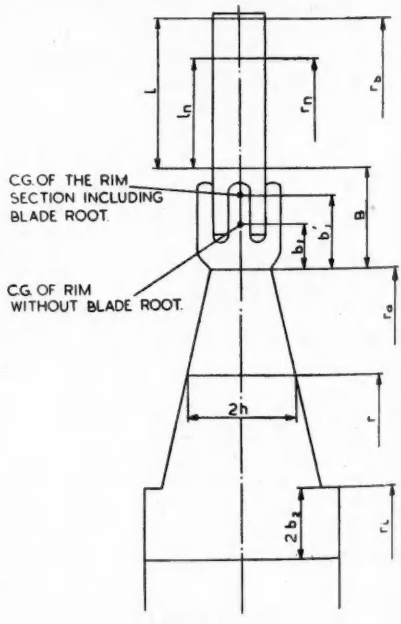


Fig. 5

section will be (cf. Timoshenko, Theory of Elasticity, Vol. II, 1916)

$$M = B\left(\frac{\psi}{r_a + b_1} - \frac{1}{(r_a + b_1)^2}\frac{d^2 v_2}{d\phi^2}\right),$$
$$B = EI_{0b}$$

where I_{0b} is the moment of inertia of the section of the disk rim (without the blade root) referred to the radial axis (see Fig. 5), v_2 is the deflection of the C.G. of the rim section b_1, the radial distance from $r = r_a$ of the disk to the C.G. (not counting the blade root), and ψ is the angle of deflection of the rim under bending, taken anti-clockwise from the initial position of the axis.

Assuming there is no radial displacement of the rim on bending of the disk, the angle of deflection of the section, ψ, is determined by the angle of slope of the disk at $r = r_a$.

From Fig. 6

$$\psi = -f'(r_a)$$

The potential energy of the bending of the rim is then

$$\Phi_\pi''' = \frac{1}{2EI_{0b}}\int_0^{2\pi} M^2 (r_a + b_1)\, d\phi$$

Substituting for M, and integrating,

$$\Phi_\pi''' = \frac{\alpha^2 E I_{0b}}{2(r_a + b_1)^3}\left\{n^2 + \left[\frac{b_1}{r_a}(n^2 - 1) - 1\right] s\right\}^2 \pi r_a^{2s}$$

The kinetic energy of the rim (including the blade roots) is determined by

$$\Phi_k''' = \frac{1}{2}\int_0^{2\pi} c^2\, dm \qquad .. \qquad (7)$$

where $c = pv_3$ is the maximum velocity of the vibration of the rim element

$$v_3 = [f(r_a) + b_1' f'(r_a)] \cos n\phi$$

b_1' is the distance from the radius $r = r_a$ of the disk to the C.G. of the cross-section of the rim, including the blade root

$$dm = \frac{\gamma}{g} F_{br} (r_a + b_1')\, d\phi$$

where F_{br} is the area of the rim section, including the blade root.

Integrating eq. (7),

$$\Phi_k''' = \frac{\gamma p^2 \alpha^2}{g}\frac{F_{br}(r_a + b_1')}{2}\left(1 + \frac{b_1'}{r_a} s\right)^2 \pi r_a^{2s}$$

BLADES.

As mentioned above, the deflection of the blade is assumed in the form

$$\beta = \frac{y(x)}{y(1)} \cos n\phi$$

where

$$\frac{y(x)}{y(1)} = \frac{\Phi(x) - x\Phi'(0) - \Phi(0)}{-\Phi'(0) - \Phi(0)}$$

The energy of the deformation of the blade element is equal to

$$d\Phi_\pi^{IV} = \frac{EI\beta^2}{2l^4}\left\{\frac{y''(x)}{y(1)}\right\}^2 \cos^2 n\phi \frac{z}{2\pi} dr\, d\phi$$

where z is the number of blades of the disk

$$dr = l dx$$

where l is the length of the blade. I is the maximum moment of inertia of the blade section.

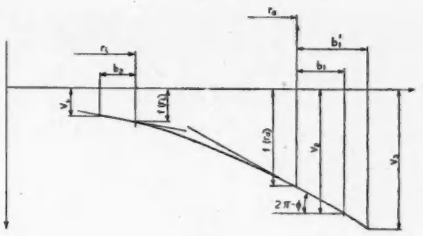


Fig. 6

Integrating with respect to x and ϕ, the energy of bending of all blades becomes,

$$\Phi_\pi^{IV} = \frac{z}{2l^3}\beta^2 \int_0^1 \frac{EI}{2}\left\{\frac{y''(x)}{y(1)}\right\}^2 dx$$

It is easy to show that

$$\int_0^1 \frac{EI}{2}\left\{\frac{y''(x)}{y(1)}\right\}^2 dx = \frac{EI_0}{-\Phi'(0) - \Phi(0)}\int_0^1 \frac{y(x)}{y(1)} dx$$

With this, the expression of the elastic energy becomes

$$\Phi_\pi^{IV} = \frac{EI_0 z\beta^2}{2[-\Phi'(0) - \Phi(0)]\, l^3}\int_0^1 \frac{y(x)}{y(1)} dx$$

while the kinetic energy is equal to

$$\Phi_\pi^{IV} = \frac{1}{2}\frac{\gamma}{g} p^2 \frac{zl}{2}\int_0^1 F v^2\, dx$$

where F is the equation of the curve bounding the cross-sectional area, and v is the amplitude of the blade deflection.

$$v = f(r_a) + (b + lx) f'(r_a) + \beta\frac{y(x)}{y(1)} =$$
$$= \alpha\left[1 + \frac{b + lx}{r_a} s\right] r_a^s + \beta\frac{y(x)}{y(1)}$$
$$0 \leqslant x \leqslant 1$$

The integrals on the right-hand sides of the expressions for Φ_π^{IV} and Φ_k^{IV} may be evaluated by formulae of approximation.

THE LACING WIRE (SHROUDING).

The kinetic energy of the wire is equal to

$$\Phi_k^{V} = \frac{\pi\gamma}{2g} p^2 F_\pi r_\pi \left\{\alpha\left[1 + \frac{b + l_\pi}{r_a} s\right] r_a^s + \beta\frac{y(l_\pi)}{y(1)}\right\}^2$$

The totalized potential energy of the disk root and disk may be expressed in the form:—

$$\pi(A_1 r_a^{2s}\alpha^2 + A_2\beta^2)$$

where A_1 and A_2 are calculated from the above formulae, whilst it is obvious that A_1 depends on s.

The sum of the kinetic energy of the disk root, disk, rim blades, and lacing wires (shrouding) may be represented in the form

$$\pi\frac{\gamma}{g} p^2 [B_1 r_a^{2s}\alpha^2 + B_2\beta^2 + 2B_3 r_a^s \alpha\beta]$$

where B_1, B_2, B_3, are again calculated from the above formulae, with B_1 and B_3 depending on s.

Equating the kinetic to the potential energy

$$p^2 = \frac{g}{\gamma} \frac{A_1 r_a^{2s} \alpha^2 + A_2 \beta^2}{B_1 r_a^{2s} \alpha^2 + B_2 \beta^2 + 2 B_3 r_a^s \alpha \beta}$$

The parameters α and β are obtained from the minimum condition for p^2:—

$$\partial p^2/\partial \alpha = 0, \quad \partial p^2/\partial \beta = 0$$

and this gives the following equations :—

$$\left(A_1 - \frac{\gamma}{g} p^2 B_1\right) r_a^s \alpha - \frac{\gamma}{g} p^2 B_3 \beta = 0$$

$$- \frac{\gamma}{g} p^2 B_3 r_a^s \alpha + \left(A_2 - \frac{\gamma}{g} p^2 B_2\right) \beta = 0$$

The condition that α and β must not vanish, leads to

$$\left(A_1 - \frac{\gamma}{g} p^2 B_1\right)\left(A_2 - \frac{\gamma}{g} p^2 B_2\right) - \frac{\gamma^2}{g^2} p^4 B_3^2 = 0$$

By selecting various values for s the corresponding values of p are obtained and the curve of p against s is plotted. The minimum value of p found from this curve will then be the required frequency.

SWEDEN

Impulse Testing of Transformers

By E. STENKVIST. (From *ASEA Journal*, Vol. 20, No. 1, January, 1947, pp. 10-16, 6 illustrations.)

THE voltages to which a transformer is subjected in impulse tests are similar in character to those which are most dangerous to transformers in actual service, *viz.* lightning surge voltages. The 50 c.p.s. tests previously recommended were inadequate in this respect since the voltage stresses thus obtained were mainly applied to the structural insulation of the transformer, i.e. the insulation between the windings and earth, whereas the strength of the internal insulation (between turns, layers and coils of the same winding) was not practically tested at all. The impulse test is the only method available that really provides a check on the internal insulation. The utility of this test depends on the possibility of detecting damage to the internal insulation, since even very small damage will reduce the dielectric strength of the transformer with respect to lightning surges, so that it may be said that no tests are preferable to unreliable tests.

Impulse testing methods were first developed in the United States. According to American standards, the tests should be made while the transformer is excited at normal voltage and supply frequency, hence the impulse voltage should be synchronized with the 50 c.p.s. voltage to ensure that the impulse voltage is impressed on the transformer winding at the moment when the system voltage across the winding in question is opposite in polarity to the impulse voltage. It has been stated that this method serves a double purpose: (1) when the impulse voltage and the system voltage are opposite in polarity, the potential differences across the windings are as large as possible, and (2) the detection of damage in the windings is facilitated since the current in the case of puncture or arcing is expected to be maintained by the system current continuing to flow at the point of fault. The latter consideration is obviously fallacious (see Fig. 1) since the voltage induced in the winding produces a current which is opposite to the impulse current at the point of fault. Consequently, if the impulse voltage is synchronized with a system voltage of an opposite polarity, this will tend to suppress the current at the point of fault. In fact, the results expected from the lagging system current are seldom obtained in practice. It would be reasonable if the impulse voltage were synchronized with a system voltage peak of the same polarity, but this would impair the usefulness of the test to a certain extent as the potential differences across the winding would in such a case be smaller than desired. For this reason, the method has not been extensively applied, but in those instances where it was used it has also given negative results regarding the effect of the lagging system current. According to the rules given in various standards, the transformer should always be tested by super-excitation, possibly at a higher frequency, after the impulse voltage tests have been made. It is exceptional that a small fault detected during impulse voltage tests is revealed by super-excitation tests. If the insulation is seriously damaged, for instance if the solid insulation is punctured, it may sometimes be possible to detect the faults in super-excitation tests. If, however, puncture has occurred in an insulation path which largely consists of oil, a fault caused by the impulse voltage test will be partially self-curing, so that the transformer will be able to withstand the super-excitation test as usual.

Therefore, if the faults occurring during the impulse voltage test cannot be found by some other method, there is the risk of their remaining latent during the subsequent tests with the result that the transformer will be delivered in a defective condition.

Other recommended indications of failures due to impulse voltage tests are sound phenomena in the transformer, evolution of smoke, or gas bubbles rising to the surface of the oil. But none of these indications can be regarded as reliable, and in fact, they are not even sufficient to be sure that the transformer is free from faults, since these indications only occur if the fault is comparatively serious, and such faults are then easier to detect by other methods.

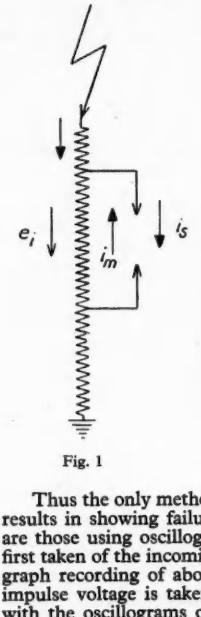


Fig. 1

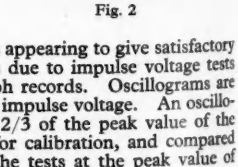


Fig. 2

Thus the only methods appearing to give satisfactory results in showing failures due to impulse voltage tests are those using oscillograph records. Oscillograms are first taken of the incoming impulse voltage. An oscillograph recording of about 2/3 of the peak value of the impulse voltage is taken for calibration, and compared with the oscillograms of the tests at the peak value of the impulse voltage. If the peak voltage recordings differ from the calibration oscillogram, this shows that the transformer winding has not withstood the test.

To distinguish small ripples in the oscillograms it is desirable that the time scale should be spread out as far as possible. For this reason it is advisable to use an oscillograph with a sine-wave time sweep (Fig. 2). The oscillographs used by ASEA have a sweep with a time amplitude of 5 microseconds per stroke. With this timing arrangement, the time markings are correct at the centre of the time scale, but are reduced to a sine scale near the turning points of the cathode-ray beam, and this affects somewhat the shape of the recorded surge wave front. The motion of the beam can generally be followed over 15 to 20 strokes (i.e. over a period of 75-100 microseconds), and detailed oscillograms with a widely spread time scale can thus be obtained. If necessary, they can be re-drawn on a linear time-base.

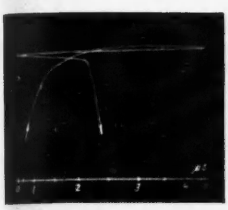


Fig. 3a

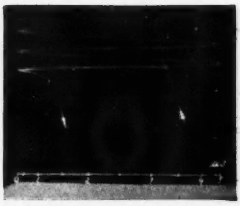


Fig. 3b

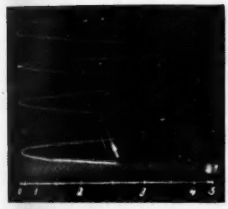


Fig. 3c

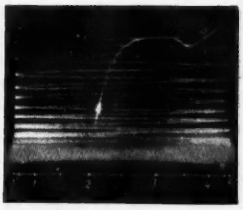


Fig. 3d

Oscillograph records of some typical faults were obtained by this method, for instance: failure due to an external flashover in air across the high-voltage bushing (Fig. 3a); external flashover in oil direct to earth (Fig. 3b); fault due to a creeping flashover along a bakelite cylinder to earth (Fig. 3c); flashover between two adjacent coils in a large transformer (Fig. 3d). In the last example, the calibration oscillogram was slightly distorted by harmonics due to the circuit constants, but the distortion of the impulse voltage wave due to the failure was so marked that the fault could be detected without difficulty. In less serious failures oscillogram distortion was less pronounced, and required more attentive examination to ensure reliable results. Difficulties increase as the output of the transformer decreases, and as its rated voltage becomes higher. Where small high-voltage distribution type transformers are concerned, this method of testing is very unreliable so that other testing methods must be used. A new method for this purpose has been advanced by J. H. Hagenguth (*Progresses in impulse testing of transformers*, A.I.E.E. Trans., Vol. 63, 1944, pp. 999-1005), who recommends replacing the line side terminal voltage oscillograms by oscillograph records of the current flowing during the impulse voltage test from the other end of the winding to earth. This method has been examined by ASEA. Oscillograms of tests made on a 260 kVA, 20 kV three-phase model transformer, which had a high-voltage winding consisting of a comparatively large number of pancake coils, clearly revealed the failures between these coils, which had been intentionally produced at several points of the winding (Fig. 4). Comparative voltage and current oscillograms of the faults described above and of the flashover at the centre of winding extending across about 2·5 per cent of the total number of turns of the winding, showed that these

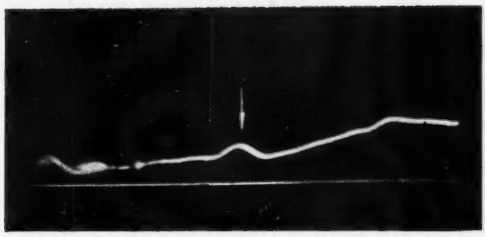


Fig. 4

oscillograms provide excellent indication of the presence of faults.

Tests on a transformer with a comparatively high rated voltage relatively to the output (100 kVA, 40 kV transformer), showed that the impulse voltage recordings were more than doubtful in this case, the distortion being scarcely perceptible, although the range of failure was 16 per cent. The current oscillograms left no room for doubt, even though considerable care was required in interpreting the results.

Further tests were made on the same transformer, with a fault extending in this case across a much smaller part of the winding (0·4 per cent of the number of turns). The fault consisted of a flashover between two adjacent layers of a multiple layer coil. In this instance, the voltage oscillograms did not provide any indication whatever, whereas the current oscillograms clearly indicated those faults which were located at the near end and the centre of the winding. On the other hand, with a flashover at the far end of the winding, the utility of the current oscillograms was rather dubious, the fault being extremely difficult to detect from these recordings.

Similar current oscillograms recorded in tests on a 20 kVA, 20 kV transformer, showed good results, while the voltage oscillograms obtained in this case did not show the slightest sign of faults.

Thus the current oscillograms suggested by Hagenguth seem to be an excellent expedient in testing low-output high-voltage transformers, while the voltage oscillograms can also be regarded as fully reliable in testing transformers of medium and high ratings. When the fault is located at the far end of the windings, the utility of current oscillograms is more than doubtful, but faults rarely occur at this point, since the front steepness of a wave which has penetrated so deeply into the winding is reduced to such an extent that the differences in voltage between adjacent layers and coils are insignificant.

CONCLUSIONS.

To ascertain whether a transformer has been damaged by impulse tests, full recordings of impulse voltages or impulse currents should be made by means of a cathode-ray oscillograph. The advisability of chopped impulse voltage tests is open to serious doubt. The impulse voltage oscillograms recorded at the line-side terminals in chopped impulse voltage tests are useless since they are cut off at the same moment as the impulse voltage collapses so that the processes taking place in the transformer cannot be followed after that moment. Attempts made by ASEA to record current oscillograms in this case have also given negative results. The collapse of the impulse voltage gives rise to high-amplitude current oscillations which predominate in the oscillograms with the result that small distortions due to partial failures are hardly perceptible. Accordingly, the impulse tests specified in the General Swedish Insulation Standards SEN.30 (exclusively confined to full-wave voltage tests) are optional, and have not been adopted as standard routine tests in view of the work involved in taking oscillograph records of each individual impulse.

BELGIUM

New Method for the Location of Faults in Overhead Transmission Lines by means of Impulse Tests

By S. MARGOULIES and P. FOURMARIER. (From *Bulletin de l'Institut Electrotechnique Montefiore*, Vol. 60, No. 5, May, 1947, pp. 123-159, 21 illustrations.)

(*Concluded from April issue.*)

EFFECT OF VARIOUS TYPES OF CONNEXIONS.

It will be convenient to treat separately the two basic problems referring (a) to the beginning, and (b) to the end point of the induced line.

(a) *Beginning of the induced line* (see Fig. 5). A line 1 with an impedance Z_1, infinitely long to the left of A, is continued from point A by a semi-infinite line 2 with an impedance Z_2 which is parallel to a semi-infinite line 2′ with an impedance Z_2' and a mutual impedance z relative to line 2. This line 2′ is also continued to infinity to the left of A by a line 1′ with an impedance Z_1', which has no effect on line 1.

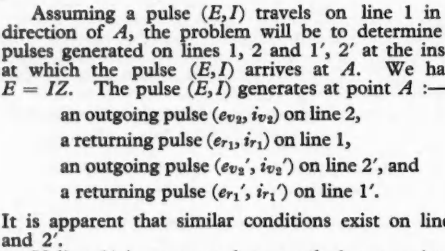


Fig. 5

Assuming a pulse (E,I) travels on line 1 in the direction of A, the problem will be to determine the pulses generated on lines 1, 2 and 1′, 2′ at the instant at which the pulse (E,I) arrives at A. We have : $E = IZ$. The pulse (E,I) generates at point A :—

an outgoing pulse (e_{v_2}, i_{v_2}) on line 2,

a returning pulse (e_{r_1}, i_{r_1}) on line 1,

an outgoing pulse (e_{v_2}', i_{v_2}') on line 2′, and

a returning pulse (e_{r_1}', i_{r_1}') on line 1′.

It is apparent that similar conditions exist on lines 2 and 2′.

If line 2′ is connected to earth by a resistance equal to Z_1', then the pulses (e_{v_2}, i_{v_2}), (e_{v_2}', i_{v_2}') and (e_{r_1}, i_{r_1}) will remain unchanged ; i_{r_1}' is then the current through the earth resistance and e_{r_1}' the voltage at its terminations.

If line 1 is replaced by a resistance $R_0 = Z_1'$ in series with an exciter apparatus schematically represented by a condenser C, then the pulses (e_{v_2}, i_{v_2}), (e_{v_2}', i_{v_2}') and (e_{r_1}', i_{r_1}') will have the same values provided we make the value of E equal to one half of the instantaneous voltage of the condenser C, and provided also that $I = E/Z_1 = E/R_0$.

The voltage at A on line 2 is then equal to e_{v_2}.

The basic equations applying to point A are as follows :

voltages on lines 1 — 2 : $E + e_{r_1} = e_{v_2}$ .. (11)

currents in line 1 — 2 : $I + i_{r_1} = i_{v_2}$ .. (12)

voltages on line 1′ — 2′ : $e_{v_2}' = e_{r_1}'$ .. (13)

currents in line 1′ — 2′ : $i_{r_2}' = i_{v_2}'$ .. .. (14)

equation of line 2 : $e_{v_2} = Z_2 i_{v_2} + z i_{v_2}'$ .. (15)

equations of line 1 : $e_{r_1} = -Z_1 i_{r_1}$ .. .. (16)

$E = Z_1 I$ .. .. (17)

equation of line 2′ : $e_{v_2}' = Z_2' i_{v_2}' + z i_{v_2}$ .. (18)

equation of line 1′ : $e_{r_1}' = -Z_1' i_{r_1}$. .. (19)

This system of equations can easily be reduced to a system with four unknowns for the currents and (with $Z_{12} = Z_1 + Z_2$, $Z_{12}' = Z_1' + Z_2'$) this gives:

$$e_{v_2} = 2E(Z_2 Z_{12}' - z^2)/(Z_{12} Z_{12}' - z^2) \quad .. \quad (20)$$

$$i_{v_2} = 2IZ_1 Z_{12}'/(Z_{12} Z_{12}' - z^2) \quad .. \quad .. \quad (21)$$

$$e_{v_2}' = e_{r_1}' = 2EZ_1' z/(Z_{12} Z_{12}' - z^2) \quad .. \quad (22)$$

$$i_{v_2}' = i_{r_1}' = -2IZ_1 z/(Z_{12} Z_{12}' - z^2) = -i_{v_2} z/Z_{12}' \quad (23)$$

$$e_{r_1} = E[(Z_2 - Z_1) Z_{12}' - z^2]/(Z_{12} Z_{12}' - z^2) \quad (24)$$

$$i_{r_1} = -I[(Z_2 - Z_1) Z_{12}' - z^2]/(Z_{12} Z_{12}' - z^2) \quad (25)$$

It will thus be seen that the conditions regarding line 2 are the same as if it had an apparent wave impedance Z_{a_2} :

$$Z_{a_2} = e_{v_2}/i_{v_2} = (Z_2 Z_{12}' - z^2)/Z_{12}' \quad .. \quad .. \quad (26)$$

Thus Z_{a_2} is independent of Z_1, and hence of the characteristics of line 1 ahead of point A, but it is a function of Z_1', i.e. of the characteristics of line 1′. If $z = 0$, then obviously $Z_{a_2} = Z_2$.

Special cases. Case 1 : $Z_1' = 0$. Line 2′ is earthed at point A (see Fig. 6). This gives the equations

$$e_{v_2} = 2E(Z_2 Z_2' - z^2)/(Z_{12} Z_2' - z^2) \quad .. \quad .. \quad (27)$$

$$i_{v_2} = 2IZ_1 Z_2'/(Z_{12} Z_2' - z^2) \quad .. \quad .. \quad .. \quad (28)$$

$$e_{v_2}' = e_{r_1}' = 0 \quad .. \quad .. \quad .. \quad .. \quad .. \quad (29)$$

$$i_{v_2}' = -2IzZ_1/(Z_{12} Z_2' - z^2) = -zi_{v_2}/Z_2' \quad (30)$$

$$Z_{a_2} = (Z_2 Z_2' - z^2)/Z_2' = Z_2 - (z^2/Z_2') \quad .. \quad (31)$$

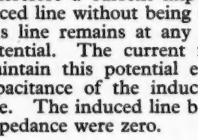


Fig. 6

Therefore a current impulse will travel along the induced line without being associated to a voltage pulse ; this line remains at any instant at a constant earth potential. The current flowing through it serves to maintain this potential equal to zero in spite of the capacitance of the induced line relative to the main line. The induced line behaves therefore as if its wave impedance were zero.

Case 2 : $Z_1' = \infty$. The line 2′ is open at A. We then have (see Fig. 7) :

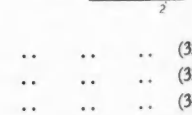


Fig. 7

$$e_{v_2} = 2EZ_2/Z_{12} \quad .. \quad .. \quad .. \quad .. \quad (32)$$

$$i_{v_2} = 2IZ_1/Z_{12} \quad .. \quad .. \quad .. \quad .. \quad .. \quad (33)$$

$$e_{v_2}' = 2Ez/Z_{12} \quad .. \quad .. \quad .. \quad .. \quad .. \quad (34)$$

$$i_{v_2}' = i_{r_1} = 0 \quad .. \quad .. \quad .. \quad .. \quad .. \quad (35)$$

$$Z_{a_2} = Z_2 \quad .. \quad .. \quad .. \quad .. \quad .. \quad (36)$$

A voltage pulse is thus propagated on the induced line without any current pulse associated to it. When the main pulse is progressing, the induced line is being

correspondingly raised to a potential determined by its capacitances relative to earth and to the conductor being tested. Positive and negative charges will then appear, resulting from transverse displacement only, without longitudinal displacement, and therefore without an apparent current. The induced line behaves, therefore, as if it had an infinite wave impedance. The apparent impedance of line 2 is equal to its natural wave impedance. In other words, the conditions are the same as if line 2′ did not exist, since it is merely raised to a potential determined by its capacities relative to line 2 and to earth.

(b) *End point of the induced line.* In the most general case, the lines 3 and 3′ have a mutual influence upon one another up to point B and are characterized by their impedances Z_3 and Z_3' and a mutual impedance z. The lines diverge at B and have beyond this point the impedances Z_4 and Z_4' (see Fig. 8).

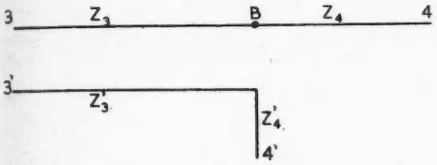


Fig. 8

In the general case, a pulse (E_3, I_3) travels on line 3 and a pulse (E_3', I_3') progresses on line 3′. This latter pulse may be due to the pulse travelling on line 3, but it may also be the result of external influences. In the first case, it only depends on the conditions at the beginning of line 3′, i.e. of the point at which line 3′ becomes parallel to line 3.

The relations (9) and (10) are valid in all cases for the pulses (E_3, I_3) and (E_3', I_3') and we, therefore, have

$$E_3 = Z_3 I_3 + zI_3' \quad .. \quad .. \quad .. \quad (37)$$

$$E_3' = Z_3' I_3' + zI_3 \quad .. \quad .. \quad .. \quad (38)$$

regardless of the way in which these pulses may have been generated.

Conditions at point B. Let

(e_{v_4}, i_{v_4}) be the outgoing pulse on line 4 ;

(e_{r_3}, i_{r_3}) the reflected pulse on line 3 ;

(e_{v_4}', i_{v_4}') the outgoing pulse on line 4′ ;

(e_{r_3}', i_{r_3}') the reflected pulse on line 3′.

We then have :

voltages on line 3 — 4 at point B: $E_3 + e_{r_3} = e_{v_4}$ (39)

currents in line 3 — 4 at B : $I_3 + i_{r_3} = i_{v_4}$ .. (40)

voltages on line 3′ — 4′ at B : $E_3' + e_{r_3}' = e_{v_4}$ (41)

currents in line 3′ — 4′ at B : $I_3' + i_{r_3}' = i_{v_4}$ (42)

equation of line 3′ : $e_{r_3}' = -Z_3' i_{r_3}' - z i_{r_3}$ .. (43)

equation of line 3 : $e_{r_3} = -Z_3 i_{r_3} - z i_{r_3}'$ .. (44)

equation of line 4 : $e_{v_4} = Z_4 i_{v_4}$ .. .. (45)

equation of line 4′ : $e_{v_4}' = Z_4' i_{v_4}'$ .. .. (46)

These eight equations with eight unknowns can be reduced to a system with four unknowns for the currents, with the result that (with $Z_{34}' = Z_3' + Z_4'$, etc.) :

$$e_{v_4} = Z_4 \frac{Z_{34}' E_3 - zE_3' + (Z_3 Z_{34}' - z^2) I_3 + zZ_4' I_3'}{Z_{34} Z_{34}' - z^2} \quad (47)$$

$$i_{v_4} = e_{v_4}/Z_4 \quad .. \quad .. \quad .. \quad .. \quad .. \quad (48)$$

These values are also those of the current leaving the point B and of the voltage at this same point B on the line 3 — 4. Furthermore :

$$e_{r_3} = \frac{(z^2 - Z_3 Z_{34}') E_3 - zZ_4 E_3' + Z_4 (Z_3 Z_{34}' - z^2) I_3 + zZ_4 Z_4' I_3'}{Z_{34} Z_{34}' - z^2} \quad (49)$$

$$i_{r_3} = \frac{Z_{34}' E_3 - zE_3 - Z_{34}' Z_4 I_3 + zZ_4' I_3'}{Z_{34} Z_{34}' - z^2} \quad .. \quad (50)$$

$$e_{v_4}' = \frac{Z_4' [Z_{34} E_3' - zE_3 + (Z_3' Z_{34} - z^2) I_3' + zZ_4 I_3]}{Z_{34} Z_{34}' - z^2} \quad (51)$$

$$i_{v_4}' = e_{v_4}'/Z_4' \quad .. \quad .. \quad .. \quad .. \quad .. \quad (52)$$

$$e_{r_3}' = \frac{(z^2 - Z_3' Z_{34}) E_3' - zZ_4' E_3 + Z_4' (Z_3' Z_{34} - z^2) I_3' + zZ_4 Z_4' I_3}{Z_{34} Z_{34}' - z^2} \quad (53)$$

$$i_{r_3}' = \frac{Z_{34} E_3' - zE_3 - Z_{34} Z_4' I_3' + zZ_4 I_3}{Z_{34} Z_{34}' - z^2} \quad .. \quad (54)$$

The symmetry of the expressions for e_{v_4}, e_{v_4}' ; i_{v_4}, i_{v_4}' ; e_{r_3}, e_{r_3}' ; and i_{r_3}, i_{r_3}' corresponds to the symmetry of the system.

Furthermore, the voltages e_{v_4} and e_{v_4}' are equal to the voltages at point B of the main and the induced line, respectively. These formulae show that, in general, the reflexions at B on either of the lines also depend on the pulses being propagated on the adjacent line. In certain cases, however, the reflexions occur in the same way as for single lines. The calculations for the general case, where the pulses (E_3, I_3) and (E_3', I_3') can assume any values, lead to the following conclusions :—

1. when one of the lines is cut off at one end, the reflexion occurs on the other line as if the first line did not exist ;
2. when both lines are earthed at one end, the reflexions occur on each of these lines as if the other line did not exist (the current is doubled in the connexion to earth) ;
3. when both lines are cut off at one end, the reflexions on either line occur as if the other did not exist (doubled voltages at one end).

These are the only cases in which the reflexions occur in the same way as on simple lines, regardless of the pulse amplitudes.

However, when the pulse on line 3′ is due only to the existence on line 3 of a pulse coming from the point at which lines 3 and 3′ converge (beginning of the parallel section) the reflexion at the end of the open line 3 is normal (doubled voltage), when the induced line 3′ is earthed at the point of convergence, no matter what the characteristics of line 4 may be, which is its continuation beyond point B.

Therefore all the other conductors should be earthed at the beginning of the line being tested, in order to have the greatest possible part of the return current flowing back through these conductors (eq. (23)); under these conditions, the pulse propagated on the test line will always be doubled at the far end of the line no matter what the connexions of the other conductors may be, and the damping can therefore be measured by merely determining, with the spherical electrodes, the pulse voltage at the beginning and the reflected voltage at the end point of the line.

TEST EQUIPMENT.

The equipment used for tests consisted of a two-stage exciter, a cathode-ray oscillograph with a special time-sweep circuit, and a potential divider. The two-stage exciter, based on a Marx circuit (see Fig. 9) had capacitances of 0·1 μF per stage, a maximum continuous charging voltage per stage of about 65 kV, and a resistance of 230 ohms between the installation and the

line. The condensers were charged by means of a potential transformer of 110/70,000 volts, a protecting resistance and a Philips Kenotron rectifier (inverse voltage 220 kV, saturation current 300 mA).

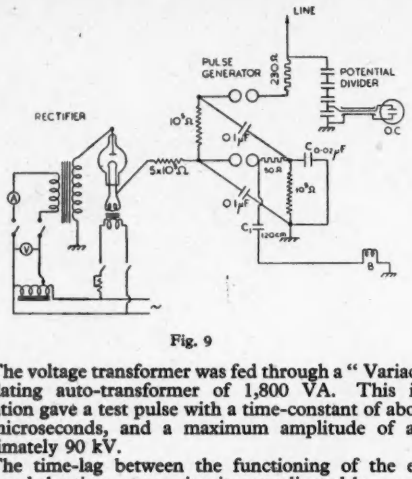


Fig. 9

The voltage transformer was fed through a " Variac " regulating auto-transformer of 1,800 VA. This installation gave a test pulse with a time-constant of about 35 microseconds, and a maximum amplitude of approximately 90 kV.

The time-lag between the functioning of the exciter and the time-sweep circuit was adjusted by means of an auxiliary condenser C and a series resistance r which retard the operation of the second stage relative to the thyratron stage which is controlled by the operation of the first exciter.

The measuring equipment consisted of a Philips cathode-ray oscillograph type G.M. 3152 with a potential divider. The resistance type divider first used was subsequently replaced by a capacitive divider of which the high-tension element consisted of two insulator supports placed one above the other, while the low-tension element was a condenser of 50,000-100,000 cm, according to the ratio desired. The timing circuit (Fig. 10) includes a thyratron T which is set off by the

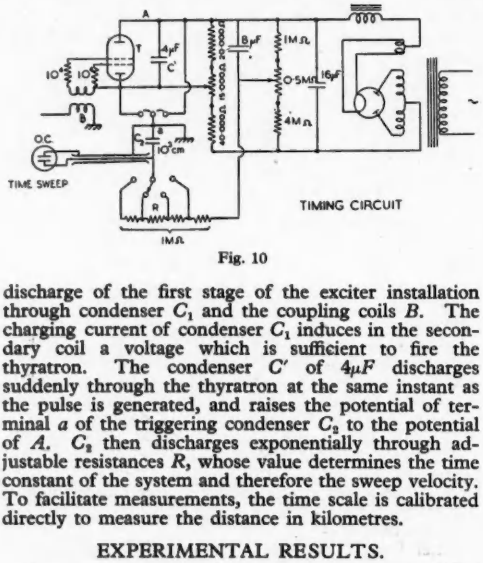


Fig. 10

discharge of the first stage of the exciter installation through condenser C_1 and the coupling coils B. The charging current of condenser C_1 induces in the secondary coil a voltage which is sufficient to fire the thyratron. The condenser C' of $4\mu F$ discharges suddenly through the thyratron at the same instant as the pulse is generated, and raises the potential of terminal a of the triggering condenser C_2 to the potential of A. C_2 then discharges exponentially through adjustable resistances R, whose value determines the time constant of the system and therefore the sweep velocity. To facilitate measurements, the time scale is calibrated directly to measure the distance in kilometres.

EXPERIMENTAL RESULTS.

The experiments were made with the object of testing the equipment and of determining the accuracy of the method under various conditions. The damping of the pulse during its travel along the line was also measured, in order to enable the determination of the minimum pulse voltage which should be used for the detection of a partial defect, regardless of the service voltage, at a specified distance from the beginning of the line on which the testing equipment is being used.

1. TESTS ON THE LINE RIMIÈRE-HAUTE SARTE (actual length 16·4 km.).

Line with one end open or fully connected to earth: The first tests were made on the middle phase wire in order to determine what type of connexions for the other wires of the line would give the clearest reflexions. The oscillograms of these tests showed that the reflexions have an optimum definition when the other two phase wires are connected to earth at the beginning of the line only, regardless of whether the end of the second phase is earthed or not. This confirms the theoretical results already obtained.

The distance measurements gave results with a scatter which was due only to the error in the time measurements, since the wave velocity is independent of the connexion of the other phase wires. Mean value for distance, measured between the origin and the first reflexion : 17·18 km. ; mean value measured from time between first and second reflexions : 16·67 km. Actual distance 16·4 km. Mean error + 5 per cent for the first, and + 1·6 per cent for the second measurement. The greater accuracy of the second measurement can be explained as follows : The brilliancy of the oscillograph was not sufficient to permit an accurate observation of the initial portion of the outgoing pulse, because the pulse is generated suddenly and has an extremely steep front, so that the first measurement had to be based on the origin of the time-sweep markings. Thus the first measurement was affected by two time-constants : (1) the time lag of the second stage of the exciter installation which is due to the auxiliary resistance-capacity circuit, and (2) the time-constant of the thyratron. If the first time-constant is the greater of the two, the sweep starts slightly before the pulse and increases the apparent time up to the first reflexion, and therefore the measured length of the line is greater than its actual value. The use of an oscillograph with greater brilliancy and if possible a photographic recording of the results would enable this error to be overcome. · Regarding the voltage divider, this can be either of the resistance or the capacity type, the second type having the advantage of greater simplicity.

Line earthed by a conductor lying on the ground over a length of a few metres. The test results were identical with those obtained when the wire was fully connected to earth.

Line connected to earth through an electrode gap: Same results as in the previous cases.

Measurement of damping during pulse propagation. For this purpose, two spherical electrodes forming a spark gap were connected at the open end of the line at Haute Sarte between the line and the earth. The voltage of the pulse-generating exciters was raised until the spark gap reached its critical value, and its functioning was observed by means of the cathode-ray oscillograph. The amplitude of the initial pulse at the beginning of the line was determined by the relations already given above, taking account of the connexions of the other phase wires, and by the charging voltage of the condensers, which was obtained by measuring the distance between the spherical electrodes of the first stage of the exciter installation or by means of an electrostatic voltmeter. These two methods gave similar results, provided the charging-time constant of

the condensers was large relative to the voltmeter constant.

If V is the charging voltage per stage, R_0 the resistance in series with the exciter installation, and Z_{a_2} the apparent impedance of the test line, the initial pulse voltage is

$$E = 2VZ_{a_2}/(R_0 + Z_{a_2}).$$

According to the theory already discussed, this pulse is reflected with twice its amplitude under the conditions of the test (beginning of the induced line connected to earth or induced line isolated at its two ends). If E_1 is the sparking voltage measured at the electrode gap, the ratio

$$\rho = E_1/2E$$

will represent what we will call the "line efficiency," that is, the ratio of the pulse amplitude at the beginning to that at the far end of the line. The results obtained were as follows :—

First case : The wires of the second line were isolated, and the two other conductors of the first line were earthed. The results were : $Z_{a_2} = 473$ ohms, $\rho = 0{\cdot}682$, $i_{v_2}' = -0{\cdot}46\, i_{v_2}$.

Second case : The five other conductors were isolated. This gave $Z_{a_2} = 541$ ohms, $\rho = 0{\cdot}615$, $i_{v_2}' = 0$.

It will be seen that in the first case the return current carried by the other wires was equal to 46 per cent of the total return current, so that 54 per cent only returned through the earth, which explains the reduction of damping relative to the case where all the wires were isolated and where all the pulse current returned through the earth.

Line tests. A further test was carried out by making a short circuit at a distance from Rimière which had not yet been determined. The measurements, based on the time-interval between the first and the second reflexion, showed an error of — 2·25 per cent over a distance of 8·77 km.

2. TESTS ON THE LINE RIMIÈRE-AUBANGE (actual distance 114 km.).

In all these tests, the wires of the second line remained in service. In these tests of a line with an open end at Aubange, and a line connected to earth at this same point, it was only possible to measure the time up to the first reflexion, because the period of the calibrated timing circuit was too short to enable the second reflexion to be observed. In this case the lack of accuracy due to the difference in the time lags of the exciter and timing circuits was less important, in view of the length of time involved in the measurement (760 microseconds). The results were as follows : distance measured from the oscillogram for the open line, 114-115 km. ; distance measured from the oscillogram for the line connected to earth, 112-115 km. Mean value, 114 km.

Measurement of pulse damping. The wires of the second line remained in service, and the method used was the same as for the measurement of damping on the line Rimière-Haute Sarte. It was found that with the two other phase wires earthed at Rimière, $\rho = 0{\cdot}51$; with the two other phases isolated at Rimière, $\rho = 0{\cdot}49$.

Line test. A test was made on a line with a short circuit at a distance from Rimière which had not been determined. The mean error of the measurements was + 0·8 per cent over a distance of 56·6 km. The fault was created (1) by connecting the wire to earth ; (2) by laying the conductor on the ground ; and (3) by inserting an electrode gap connected to earth. In all three cases identical results were obtained.

Pulse amplitude measurements showed a damping $\rho = 0{\cdot}61$ over this distance. The voltage dropped by about 39 per cent over the first half of the line, and by about 10 per cent only on the second half. This is probably because the decrease in steepness of the front of the pulse is much more pronounced on steeper waves, i.e. during the first kilometres of the course.

MODIFIED METHOD : LINE TESTED WITH CONTINUOUS VOLTAGE.

In cases where the fault was situated near the far end of the line, on account of damping and in order to observe the successive reflexions, it was found necessary to insert a resistance in series with the exciter installation. The total charging voltage for the condensers which was then required was considerably higher than the usual test voltage.

To avoid this drawback, the method was modified as follows : The line was isolated by cutting out the voltage transformers, and then tested with a continuous voltage supplied by a transformer, a kenotron rectifier and a protecting resistance of a few hundred thousand ohms.

When the charging voltage U reaches the breakdown voltage of the fault, a rectangular pulse due to the discharge travels from the point of fault M towards the beginning of the line A and is reflected there ; the returning pulse is again reflected at M, and then returns to A, etc. A cathode-ray oscillograph connected at A will show therefore a series of voltage pulses separated from each other by a time interval $2D/v$, the value of which enables the distance D to be determined. The time-sweep system is triggered by the first discharge pulse arriving at A. Owing to the damping on the line, only the first pulses are rectangular, and the subsequent pulses acquire a quasi-sinusoidal shape.

This method was tested by measuring the leakage resistances of high-tension cables. The first tests were made on the line Rimière-Haute Sarte on three different days, during rainy, foggy, and very rainy weather. At the beginning of these tests the insulation resistance was 2·47, 1·78 and 2·06 megohms, respectively. The insulation resistance increased during the tests, owing to the gradual drying effect of the leakage current, and the results are summarized in the following table :—

Test No. Weather	1 rainy	2 foggy	3 very rainy
Final charging voltage	78 kV	74·7	70·7
Leakage current ..	10·5 mA	10·1	17·8
Insulation resistance	7·4 megohms	7·31	3·98

The weather conditions during test 3 can be regarded as about the most unfavourable that can be encountered in practice ; it therefore seemed reasonable to assume for the minimum value of the insulation resistance after the voltage had been applied for about half an hour, a value of 4 megohms, or 66 megohms per kilometre.

Further tests on the line Rimière-Aubange. A single test was made in foggy weather. Results obtained after applying the voltage for about 20 minutes : final charging voltage, 54·3 kV ; current, 17 mA ; insulation resistance, 3·2 megohms. The insulation resistance was thus 365 megohms per kilometre.

ADDITIONAL TESTS.

Since the testing equipment was put into service, three faults occurred on lines ending at the Rimière station :—

(1) The first fault, on the line Rimière-Haute Sarte occurred in December, 1946 ; the main line switch at Rimière was frozen and did not function, so that the entire network was thrown out of action. The two parallel lines were tested, and the fault points were determined with an accuracy between 0·13 and 1·8 per cent.

(2) Fault on the line Rimière-Haute Sarte, one phase wire earthed. Position determined with an error of – 1·1 per cent.

(3) Fault on the line Rimière-Farciennes. This was an intermittent fault, which was only noticeable in small proportion of the discharges (about one in fifty), the line appearing to be in good condition during most of the time. The measurements were therefore more difficult. Measured distance 34·5 km, actual distance 33·5 km, error + 1 km, or + 2·4 per cent. An inspection of the fault showed that it was due to the earth cable which was cut near the protecting ring of the defective phase wire, thus producing intermittent discharges.

CONCLUSIONS.

Impulse-testing method. By observing the successive reflexions of the pulse, the position and the type of the fault can be determined with an accuracy of 1 — 2 per cent in the distance. The accuracy could be increased by taking photographic recordings and using an oscillograph with greater brilliancy.

Pulse damping during travel is not excessive, and damaged insulators can be passed even when the fault is situated at a considerable distance, without resorting to extremely high pulse voltages.

When the breakdown voltage of the fault is equal to $U_n \sqrt{2}$ (where U_n is the effective service voltage per phase), the line will not be able to withstand the service voltage. However, it will be an advantage to be able to find the position of a fault which may have a slightly higher breakdown voltage. If a safety coefficient of 1·15 is assumed, the fault which has to be located will have a breakdown voltage of $1{\cdot}15\ U_n \sqrt{2}$ at 50 cycles/sec.

Furthermore, if K is the impulse factor of the fault for the type of pulses being used, and ρ the efficiency of the line up to the point of fault, then the pulse voltage required will be

$$E = 1{\cdot}15\, K U_n \sqrt{2}/\rho,$$

provided the length of the line beyond the fault is sufficiently great relative to the length of the pulse, so that the reflexion at its extremity will not be superimposed to any appreciable extent upon the outgoing pulse. Assuming that $K = 1{\cdot}15$, we find

$$E = 1{\cdot}32 \sqrt{2}\ U_n/\rho = 1{\cdot}86\, U_n/\rho.$$

Furthermore, let R_0 denote the resistance between the exciter equipment and the line, and Z_{a_2} the apparent impedance of this line ; then the total charging voltage of the exciter equipment which is required to pass over the fault is given by

$$U_0 = 1{\cdot}86\,(R_0 + Z_{a_2})\, U_n/(Z_{a_2}\rho).$$

With $R_0 = 230$ ohms, and $Z_{a_2} = 470$ ohms (mean value) we have $\rho U_0 = 2{\cdot}76\ U_n$. For a 70 kV network $U_n = 41$ kV., and therefore $U_0 = 114$ kV. Actually the line is limited in length, and the reflected pulse is superimposed on the incident pulse and appreciably increases the voltage at the fault, when this is situated near one end of the line.

In the first approximation, it will be assumed that the pulse retains its shape, but decreases exponentially in amplitude with the distance travelled. Its amplitude is then given by

$$e = E \exp(-\alpha x),$$

where x is the distance from the exciter equipment to the point being considered, and α the damping constant determined experimentally in a previous test.

The variation of the voltage of the outgoing pulse at a point M as a function of time is therefore :

$$e = E \exp\left[-\left(\alpha x + \frac{t}{T}\right)\right]$$

where T is the time-constant of the tail of the wave, while $t = 0$ corresponds to the instant at which the front of the pulse arrives at the point of fault M.

The reflected pulse arrives at the same point M after a time $2d/v$ after the first outgoing pulse, d being the distance between the point M and the end of the line, and v the wave velocity. This reflected pulse produces therefore at point M a voltage

$$e_r = E \exp\left[-\alpha\,(x + 2d) + \frac{1}{T}\left(\frac{2d}{v} - t\right)\right]$$

The resultant voltage is therefore

$$e' = E \exp\left(-\alpha x - \frac{t}{T}\right).\left[1 + \exp\left(-2d\alpha + \frac{2d}{vT}\right)\right]$$

This expression is valid on for $t > (2d/v)$. For $t < (2d/v)$ the voltage at M is given by the outgoing pulse, i.e. by $e = E \exp[-\alpha x - (t/T)]$. The greatest value of the voltage at M is attained when $t = 2d/v$, at the instant when the reflected pulse reaches M, this value being given by

$$e'_{max} = E\left[\exp\left(-\alpha x - \frac{2d}{vT}\right) + \exp(-\alpha x - 2d\alpha)\right],$$

while the first outgoing pulse has at point M the amplitude $E \exp(-\alpha x)$. The maximum voltage at M is therefore the greater of these two values. The voltage due to the superposition of the initial pulse and the reflected pulse will be greater than the voltage due to the initial pulse alone, only if $[\exp(-2d/vT) + \exp(-2d\alpha)] > 1$, and this condition determines the maximum value of d.

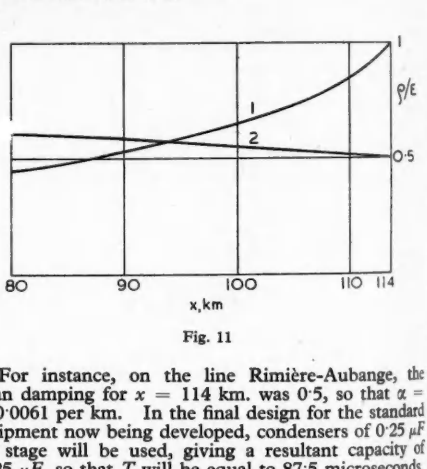


Fig. 11

For instance, on the line Rimière-Aubange, the mean damping for $x = 114$ km. was 0·5, so that $\alpha = 0{\cdot}0061$ per km. In the final design for the standard equipment now being developed, condensers of 0·25 μF per stage will be used, giving a resultant capacity of 0·125 μF, so that T will be equal to 87·5 microseconds. Fig. 11 gives the curves of the maximum voltage due to the initial pulse alone (curve 2) and of the maximum voltage resulting from the superposition of the initial and the reflected pulse (curve 1). These curves show that for distances exceeding 94 km. the superposition of two pulses gives a voltage which is greater than that of the incident pulse alone. The minimum value at $x = 94$ km. is 0·56, therefore $U_0 = 114/0{\cdot}56 = 202$ kV.

The test equipment was built from material available in University laboratories and in the stocks of the Linalux. Portable equipment with low-consumption elements is now being developed. The tests can be made by two men (one operator and one assistant). The testing of the three phases of a line can be done in 20 minutes, while a double line with six phases can be tested in 30 minutes.

The 500-kV Tidd Test Project

(From *Electrical Engineering*, Vol. 66, No. 12, December, 1947, pp. 1178-1181 and 1234-1235, 5 illustrations.)

RECOGNISING the need for additional information on the characteristics and performance of materials and equipment constituting transmission systems at very high voltages a number of large manufacturers decided to cooperate in an extensive research and test programme. Three full-sized test lines with associated equipment designed to operate at 500 kV were constructed, and several companies are participating in the high voltage tests.

Three parallel lines constitute the transmission system. They are connected directly to a bus at the test station without circuit breakers and are fed through a 5000 kVA bank of three single-phase transformers. Three single pole arresters have been installed to protect the transformers. An oil circuit breaker, protected by an arrester, will be connected to the bus. Measuring instruments are mounted on the top of the transformer bushing and will be read by telescope from an observation booth some 50 feet distant.

Within the test station all bus work is of 2 inch segmental hollow conductor. Shielded measuring leads are enclosed in the hollow conductors and extend from the instruments in the housing box on top of the high voltage transformer bushing to the entrance of each of the three lines.

Two of the three test lines are 1·4 miles long, and the third is a single 800 ft. span. The two longer lines have an average span of about 1050 feet and comprise seven towers per line. The lines run approximately parallel and about 288 ft. apart. The conductors are approximately 85 ft. above ground at the towers and about 40 ft. at midspan. The ground wires extend over the station for bus and equipment shielding. The lines are phased so that adjacent conductors of different lines connect to the same phase.

Transformers. Three single-phase transformers are used to step up the 66 kV supply to high voltage that can be varied between 264 and 500 kV between phases. The 66 kV side is delta-connected, the 500 kV side is connected in star with the neutral solidly earthed. The impedance is 6·6 per cent which is comparatively low, but of advantage in maintaining a sinusoidal wave shape.

The windings are arranged in the manner normally used in shell type high voltage transformers. The large tap range was obtained by using a 10 per cent tap range in the high voltage winding and a large number of taps in the low voltage winding. The no-load tap changers are operated from hand wheels on the tank wall. The total tap range is covered in 24 steps.

A unique feature is the provision for a high voltage metering winding at high-voltage line potential. This is obtained by extending the high-voltage winding for a few turns and bringing the lead through the bushing with the main line lead. This portion of the winding and its lead are electrostatically shielded so that it does not carry appreciable capacity current. This winding provides a voltage for metering and power for the other equipment mounted in the cabinet at the top of the high voltage bushing. In addition, the conventional metering windings are provided at a low-voltage level.

The core and coil assembly of the transformers is built into a form-fit tank, the two sections of which were welded together at the factory. This type of construction made possible the shipment of the unit complete in oil by laying each transformer on its side on a flat car.

Lightning Arresters. Two different types of lightning arresters have been installed. These are on either side of the observation booth. Both are electrically alike and for a maximum voltage rating of 350 kV. One

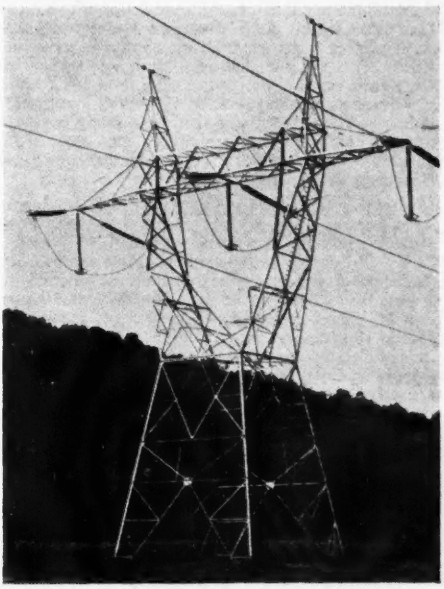


Fig. 1. One of the line towers showing the insulator and ground wire arrangement.

arrester is designed for suspension from a steel structure. The units are suspended in zig-zag fashion between strings of insulators. This shortens the length considerably. The second type is self-supporting. There are advantages and disadvantages in each. Grading rings are used to control the electrostatic field. The units are standard units as used in auto-valve station-type arresters of all commercial ratings.

Line Towers and Sub-Station Structures. The line towers (Fig. 1) support three conductors, 100 ft. above ground at the point of attachment of the insulator strings, and two ground wires located 24 ft. above the conductor cross-arm. The conductor cross arm is 90 ft. long providing a maximum phase spacing of 45 ft. Attachments are provided for variable conductor spacing of 45, 38½ and 32 ft. between the conductors. Alternate attachment points for the ground wires are provided so that the wires can be lowered 6 or 12 ft. below the normal topmost location.

The sub-station structures are essentially similar to the line towers except that the height and cross-arm length are less. There is also one cross girder structure which supports two suspended lightning arresters in addition to the ground wires and the busses. It also has attached to it the steel frame observation and control booth.

Insulators and Station Hardware. The lines are strung with 30 conventional 5¾ inch space insulators per suspension string. Provision is being made for determining the effect of shorter strings. Different types of grading rings are provided and in all cases are connected to the line end of the string only. Special hardware for each particular type of conductor is provided. A new type of suspension clamp was developed which is about 40 per cent shorter than the conventional types. The corona shield, involving a novel means of support completely shields the hardware at

the lower end of the insulator string. Tests have proved that the 2 inch diameter material used for the corona shield gives adequate corona suppression.

Conductors. The line conductors now installed are 1·65 inch outside diameter, segmental copper conductors on one line ; 2·0 inch diameter aluminium cable, steel reinforced with paper filler, on the second line ; and 1·65 inch outside diameter stranded copper with I-beam core on the short line. In addition to these single conductors, the so-called " bundle," or grouped conductor arrangement, will be tested, using individual conductors ranging in outside diameter from 0·7 to 0·92 inch, using from 2 to 4 conductors per phase, and spacings of 6 to 18 inches.

Switchgear Equipment. The single-phase, low oil content impulse type circuit breaker was designed for a 3-phase interrupting rating of 10,000,000 kVA at 360 kV. To obtain high speed reclosing a compressed air mechanism was designed, so that the operation of each pole by its own mechanism permits either 3-phase or single phase reclosing. The circuit breaker can reclose in 10 cycles or less, if that should prove practical from a dielectric recovery standpoint of the arc path. The large shielding rings at either end of the horizontal interrupting tube divide the voltage uniformly between the breaks.

The arresters installed to protect the switchgear equipment consist of 25 standard station-type thyrite interchangeable 12 kV units, and have a maximum permissible line-to-ground rating of 300 kV.

Instrumentation and Measurement. Atmospheric conditions, radio influence, and corona loss measurement are required with great accuracy. A new test method was, therefore, developed and the circuit was designed on the following lines :

1. Corona power to be measured by a wattmeter located at a point of high potential to eliminate necessity of subtracting insulating transformer losses.
2. Wattmeter potential to be taken from an extension of the transformer high-voltage winding.
3. Automatic compensation to be provided to correct for the phase angle errors in the potential as applied to the wattmeter.

The system required the development of a special wattmeter which is about 100 times as sensitive as an ordinary wattmeter ; the current coils are in series with the line. Measurements can be recorded over long periods of time ; slow-moving meters integrate effects over long periods of time and a high-speed meter permits study and co-ordination of loss under abnormal weather conditions. An instrument for measuring ground gradients which may influence corona is also provided. An important part of the tests will be the measurement of radio noise originating as a result of corona.

RUSSIA

Rapid Method of Moisture Determination in Coals by High Frequency Currents

By I. L. Liplavk. (From *Zavodskaya Laboratoria*, No. 6, 1946, pp. 555-558, 3 illustrations.)

Methods so far suggested for rapidly determining the moisture content in coals have failed to satisfy requirements. This was due, in some cases, to the difficulty of establishing and maintaining proper test conditions, as for instance in the dielectric constant method, and because other methods only proved reliable when applied to certain types of coal. The exothermic method is one example demonstrating this kind of unreliability, which is not due to imperfect development but is simply a result of using an underlying principle not generally applicable to all types of coal and for this reason its usefulness is related to coals mined in certain districts only. As drying the coal is still the most convenient and suitable procedure, the problem was to convert it into a rapid method whilst fully conserving its accuracy.

The idea that high-frequency heating would effect this purpose was suggested by the fact that the temperature rise in a high-frequency field is almost instantaneous and simultaneous throughout the heated mass. It is known that this results in the interior of the mass becoming hotter than the surface, because the latter dissipates heat in the ambient air. As the temperature rise depends not only on the energy and frequency of the field, but also on the properties of the medium in which it is set up, the effect is selective. This peculiarity of the high-frequency field is utilized for heating the water content, whilst having less effect on the solid coal the temperature rise of which is due mainly to heat conduction. On the other hand, this principle materially changes the rapidity and completeness of the drying process, whose results are further assisted by the outward temperature gradient which also drives the moisture outwards. Because the ambient air is not heated up by the field, much higher temperatures may be used for the purpose of drying than is possible by any other method.

The high-frequency field is produced in the usual way by a self-excited valve oscillator inductively coupled to the oscillating circuit into which the specimen to be tested is brought, i.e., it is inserted between the plates of a variable condenser, which permits rapid changes of the field density and provides a convenient means for controlling the temperature rise within fairly wide limits. The condenser plates are of copper mesh to allow the steam formed to escape.

As the working voltage is 4,000 volts, care was taken to incase the whole device and make the internal parts inaccessible whilst under pressure. The whole process is more or less automatic and the necessary controls are arranged outside the metal chassis of the apparatus. The weighing of the coal sample, or samples, because two small lumps of about 5 gr. weight are used, is carried out by a torsion type balance, on which the specimens are suspended in silken nets. One of these nets also carries a thermometer, which is lowered between the condenser plates, and thereafter the net carrying the second specimen is similarly lowered. When the door of the apparatus is closed, the heating current and anode voltage are switched on. To accelerate the process, the condenser plates are initially in contact with the samples, but after reaching a temperature of 120 to 130 deg. C. they are taken back far enough to maintain a constant temperature. The torsion balance then allows a reading of the loss in weight in milligrams to be taken. This is done immediately the balance has settled down. The weight of the sample with the thermometer is taken last, and thus a control reading is obtained. The whole procedure, if carried out by skilled personnel, should not take more than 2 min. The accuracy of the method is equal to that of the distillation method which takes as long as 2 hours. It should be noted that the given measuring time applies to air-dry coal, but even in the case of a very moist coal the time including weighing need not exceed 10 minutes.

CLASSIFIED ABSTRACTS

Numerous important abstracts which, for lack of space, cannot be published in our main editorial columns, are regularly included in condensed form in this section. Subscribers may obtain photostat copies of all original articles at cost.

ELECTRIC BATTERIES

Emergency Supply Systems with Accumulator Batteries

By H. A. W. KLINKHAMER. (From *Philips Technical Review, Holland*, Vol. 9, No. 8, 1947/48, pp. 231-238, 10 illustrations.)

EMERGENCY supply systems are for serving electric mains-fed plant, such as telephone or lighting installations, in the event of a breakdown in the mains supply. A very suitable system is one where the plant is fed from a rectifier with a battery of lead-cell accumulators connected to it in parallel. The rectifier has to be of such a construction as to ensure a constant output voltage practically independent of mains voltage fluctuations and variations of load. This is necessary to maintain a permanent charge of 2·1/2·2 V per cell in the battery, which modern experience proves to be the condition ensuring the longest life for a battery. A rectifier answering these requirements is therefore called a " preserving rectifier." This article explains the advantages of such an emergency supply system compared with a dynamo and the obsolete two-battery system (with two batteries feeding the plant alternately). Finally it is shown that a type of rectifier already described in this journal*, with a highly saturated transformer, is quite suitable as a " preserving rectifier," two special applications of which for emergency supply systems are described.

* *Philips Techn. Rev.*, Vol. 6, 1941, pp. 39-45.

ELECTRICITY AND MAGNETISM

Registering Fluxmeters.

By G. BATTISTINI. (From *L'Elettrotecnica, Italy*, Vol. 35, No. 1, January, 1948, pp. 11-20.)

THIS article gives the theory of an instrument capable of measuring the instantaneous magnetic flux $\Phi(t) = \int_0^t u(t)\,dt$. The instrument is a pivoted coil which is deflected by the external magnetic field. By means of a number of coils of this type, situated at the seabottom at the entrance of harbours or channels, it is possible to determine with sufficient accuracy the magnetic field around the hull of a battleship. Results are obtained much more rapidly than with the usual point-to-point methods.

After discussing the conditions for an ideal instrument of negligible inertia and with a damping torque equal to zero, the author investigates the conditions in the practical case and determines the optimum inertia and damping constants for measurement errors of 1 and 2 per cent. In this analysis, the induced e.m.f. is resolved into its sine components, and the amplitude and phase errors, as well as the error for the continuous component, are determined for the steady and the transient state. The natural frequency of the instrument should be chosen as near as possible to the geometric mean of the extreme frequencies which are to be registered, while being given a high value in order to limit the error in registering the continuous component. Tables of damping coefficients, and the numerical constants for an instrument taken as an example, are also given. The instrument has a sensitivity of about 700 Maxwell/degree.

FOUNDRY PRACTICE

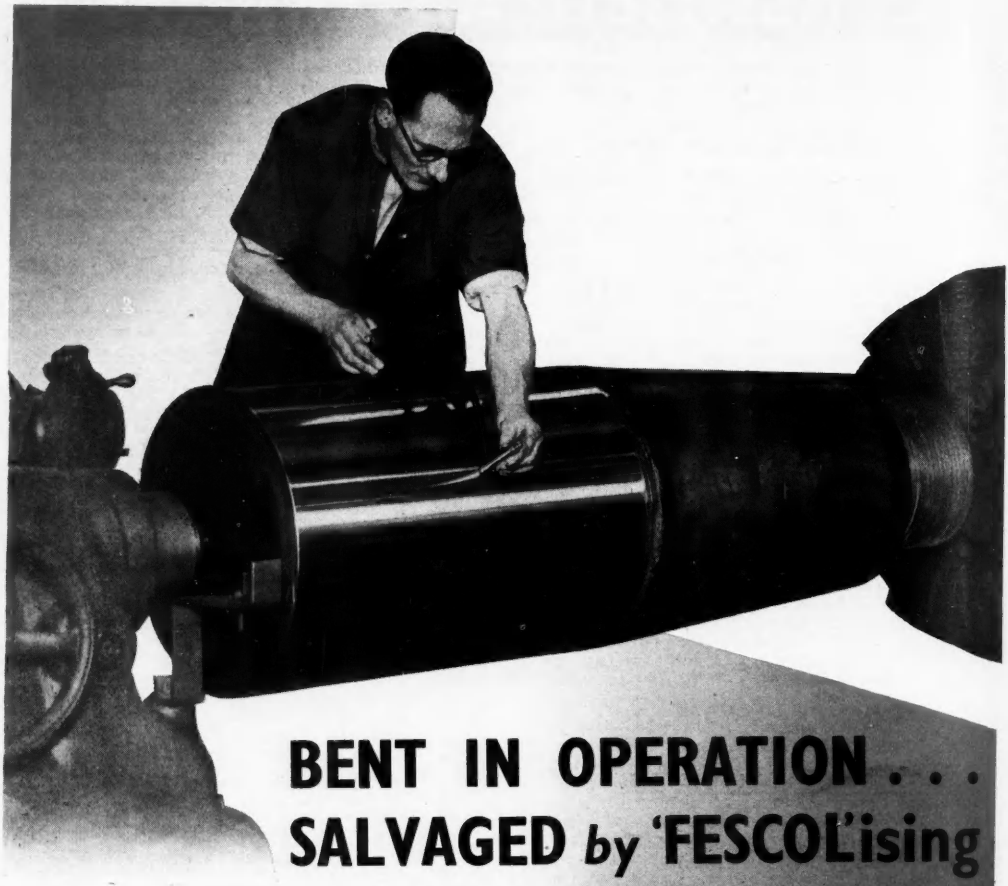
Development and Present Technique of Foundry Facing Materials.

By P. NICOLAS. (From *Fonderie, France*, Vol. 22, October, 1947, pp. 843-852, 7 tables.)

FACING materials are either added to the moulding sand, or the mould and core surfaces are coated, in order to produce a smooth casting surface and prevent reaction between the sand and the molten metal or vitrification of the sand. The materials used were " blacking " or " black-wash " because their main constituent was graphite, but now different facing materials are also being used, especially in the U.S.A.

Mineral black, with powdered carbon as a base, is added to green sand in amounts of 3 to 5 per cent. It burns on contact with the molten metal and the gaseous products of combustion form an insulation between the sand and the metal. German and U.S.A. practice makes use of a material having 35 per cent volatile matter, 58 per cent free carbon, and up to 7 per cent ashes which gives excellent results. The American material can also be obtained in different grain sizes and thus is adaptable to the sand chosen. In Great Britain a satisfactory material with high carbon, medium ash, and low volatile matter content is used. The ash and volatile matter contents are varied according to the weight of the casting. Special mineral mixtures are used for magnesium castings to give a neutral protecting atmosphere to the metal. Vegetable charcoal, added in amounts of 3 to 5 per cent of the sand, produces a softer combustion reaction and is used for delicate castings where the slight chilling effect of mineral black is undesirable. The best material is produced from soft wood, charred at 450 to 500 deg. C., and crushed. Volatile matter should not exceed 5 per cent, nor ashes 10 per cent of the material. A special facing material for pattern sand is recommended in the U.S.A., which contains 76 per cent volatile matter which consists essentially of wheat meal. Sometimes, a fine siliceous core sand with a linseed oil binder is used as a facing sand. It liberates hardly any gaseous products, and its permeability need not therefore be great. A smooth casting skin can thus be obtained.

" Wet-black," a complex mixture containing graphite coke, vegetable charcoal, and clay is brushed onto the surfaces of the mould and core. The German product " Ariston " is known for its superior quality; Ceylon plumbago, used in the U.S.A., has very similar chemical constituents (6 per cent volatile matter, 62 per cent free carbon, and 32 per cent ashes). Belgian and English plumbago have more carbon, less ashes, and give generally satisfactory results. Coating technique is well developed in U.S.A. where the coating material (50 per cent mineral black, 45 per cent clay, and 5 per cent ochre for cast iron, and 75 per cent powdered silica, 20 per cent clay, 5 per cent ochre for cast steel) is torch dried for 30 minutes. Powdered silica endangers the health of workers and is often replaced by ground chamotte. Facing materials such as graphite, and talc, can be dusted on, or sprayed on (petrol). In U.S.A. practice, Ceylon plumbago is suspended in benzol,



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alcohol, or paraffin, and sprayed on. Another mineral refractory material (about 50 per cent clay, 25 per cent talc, and 25 per cent ochre) is slaked on the mould surface and permits castings of up to 1100 lbs. to be cast in green sand. Due to its alcalinity, it facilitates the enamelling of the casting surface where this is required. Facings that are to be sprayed on must be carefully prepared, dry grain size as well as viscosity of the mixture must be controlled, very pure calcium-free water or other liquid carriers must be used and the mixture must be continuously stirred. The right method of application is equally important.

HEAT TREATMENT

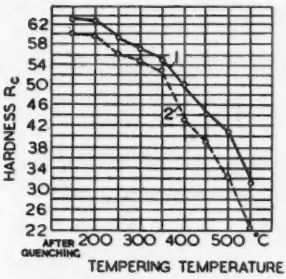
Cyaniding Paste for Tools.

By V. A. IVANOV. (From *Vestnik Machinostroenia, Russia*, No. 6, 1947, pp. 71-72, 1 illustration.)

THE Central Laboratory of the Kolomna Locomotive works has developed a new method for the cyaniding of tools with the use of a paste. This method consists in covering the part of the tool to be treated with a 6-8 mm. thick layer of a special paste. After the paste has dried, the tool is packed in a box and covered with iron filings ; it is then placed in the furnace.

The paste consists of 50 per cent powdered charcoal, 25 per cent sodium chloride and 25 per cent potassium cyanide. These constituents are carefully mixed and water is added to make the paste have a water content of 15 per cent. This process may be applied to both plain carbon and alloy steel tools. Treatment is carried out at 780-820 deg. C. with subsequent quenching in water. The length of the treatment is two hours.

This method was employed at the aforementioned laboratory for the cyaniding of 0·8 per cent carbon steel. In this case the length of the treatment was three hours at 820 deg. C. After quenching, all test pieces were tempered, the temperatures ranging from 200 to 500 deg. C. (in intervals of 50 deg. C.). One of the test pieces was allowed to cool while remaining in the box. The Rockwell hardness values obtained are charted in the accompanying diagram as a function of the tempering temperature. The thickness of the cyanided zone was found to be 0·1–0·15 mm. In practice this method is applied to tools of a steel with 0·85–0·95 per cent carbon, 4·0–4·5 per cent chromium, 8·5–9·5 per cent tungsten, 2·0–2·6 per cent vanadium. The tools which are in boxes are then heated to 550 deg. C. in the furnace and are held at this temperature for two hours; they are subsequently quenched in water.



(1) cyanided. (2) not cyanided.

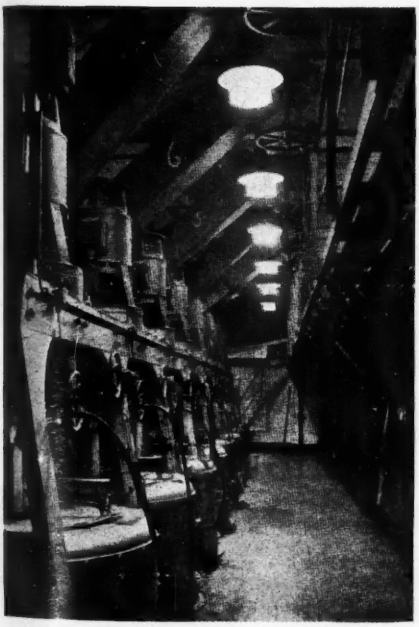


Fig. 1. **Composite lighting in a sugar refinery.**

ILLUMINATING ENGINEERING

Illumination in Factories by Composite Light Sources.

By G. WILLHEIM. (From *Elektrotechnika, Hungary*, Vol. 39, No. 10, October, 1947, pp. 194-195, 1 illustration.)

FOR factory illumination, use can be made of the following types of light sources: (1) the incandescent lamp ; (2) the vapour lamp ; (3) the fluorescent tube ; and (4) the composite light source (combined incandescent and vapour lamps).

For composite lighting the vapour lamp used should be of the mercury vapour type, and this, in conjunction with the incandescent lamp, will give a colour similar to sunlight. The mercury vapour lamp is of the usual type, with liquid mercury evaporated by the heat of an electric current, and two electrodes between which the luminous gas discharge takes place. The lamp is made up of two concentric glass tubes ; the outer tube serves as a heat insulator, a vacuum of about 0.1 atmosphere being maintained between the two tubes. For the same candle-power, mercury vapour lamps of type 33/45 lm/w operate at a lower wattage than incandescent lamps.

Fig. 1 shows the composite type of lighting used in a sugar refinery. Each lamp consists of a 300-watt mercury vapour element and a 200-watt incandescent lamp, with a total power of 6,000 lumen. The mercury vapour emits only the lines 404·7, 435·8, 546·1 and 577 in the violet, blue, green and yellow regions of the solar spectrum, whilst the incandescent lamp supplies the missing red light without which there would be a colour distortion. The intensity of illumination is 80 lux.

If a sodium-vapour lamp is used instead of the mercury vapour lamp, the composition of the light is modified since sodium emits the line 589 which is a pure yellow. This sodium light blended with the incandescent light will still therefore give colour distortion, so that its use has to be limited to applications where colour identification is not important, as for instance in mines, tramways, etc. Artificial daylight can be obtained by introducing fluorescent materials into the space between the inner and the outer tubes of a mercury vapour lamp.

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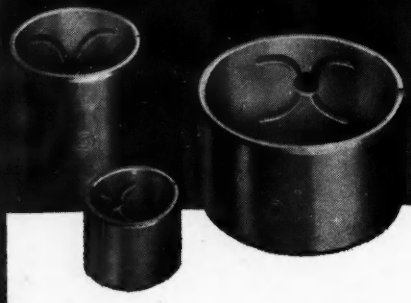

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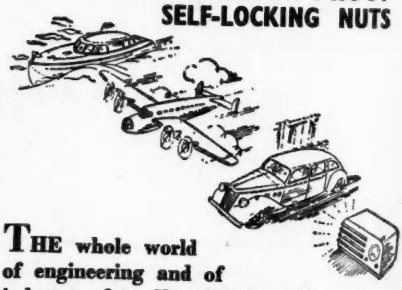

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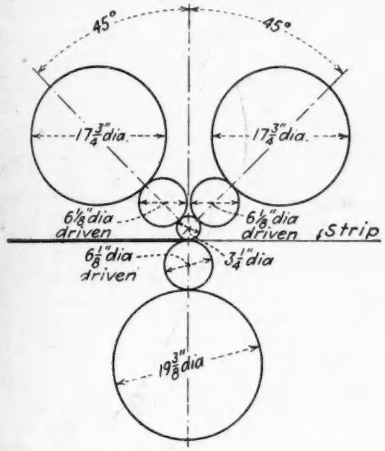
ROLLING MILL PRACTICE

The Cold Strip Y-Mill

By A. B. MONTGOMERY and W. M. MCCONNELL (From *Iron and Steel Engineer, U.S.A.*, Vol. 25, No. 3, March, 1948, pp. 99-104, 8 illustrations.)

A NEW type rolling mill, called a Y-mill, the rolls of which are in the form of a Y-shaped cluster, permit the rolling of very thin strip as well as heavier gauges in both low and high carbon and stainless steels.

Due to the fact that the housing windows are very short, and the roll pressure light, there is practically no stretch in the housing when taking maximum reductions. The fundamental design of the cluster part of the mill is to provide means for maintaining an approximate 90 deg. angle from the centre of the small friction driven baby or top work roll through the axles of the intermediate back-up or driven rolls and the large main top back-up rolls. This arrangement is maintained by shims under the top back-up roll chocks, the shims being used to compensate for roll grinding and



for maintaining correct alignment, of the top roll assembly. The rolls of the mill are arranged in the form of the letter Y, as indicated in the accompanying illustration.

Each upper back-up roll is clamped in place at the top of its housing window, with a self aligning rocker plate, to provide protection for the roll neck bearings against roll deflection. The two upper intermediate rolls and baby roll are balanced by springs or hydraulic cylinders, mounted in the lower work roll chocks to maintain roll contact between all the rolls when idling the mill.

The 6½ in. dia. lower work roll and the two 6½ dia. upper intermediate rolls are driven through spindles from the mill pinion stand. The diameters of the three driven rolls are matched to the normal variation in diameter allowable in four-high mills. The baby work roll and the back-up rolls are not driven. The mill can be arranged with the driven rolls from 5⅛ to 7¼ in. and the baby roll from 2⅛ to 4 in.

The lower work roll and the back-up roll are adjusted up and down on non-rotating housing screws, mounted in the lower part of the housing windows. All screw-down parts are designed so that the rolls may be forced into the strip. A single 5 h.p. motor, with reduction gears, furnishes power through a gear type mechanical clutch to each screw.

The coilers are mechanically collapsed, or expanded, by a new extremely simple and rigid method. When expanded, the reels grip the strip for rotation in either direction. The motors driving the reels produce either pull or drag by acting alternately as motors or generators. Sufficient tension is produced by the reel drives so that the strip will be rolled flat and tight on the coil.

TRANSFORMERS

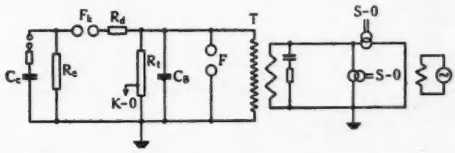
The Impulse Strength of Transformers.

By F. BELDI. (From *Brown Boveri Mitteilungen, Switzerland*, Vol. 34, No. 6/7, June/July, 1947, pp. 105-115, 12 illustrations.)

THIS article describes impulse tests on two transformers of 24 and 160 kVA rated power, designed for an operating voltage of 8,000 volts, and further tests on two other transformers of 2,000 and 5,000 kVA power at 45,000 volts. The transformers had been in operation for nearly twenty years without giving any trouble and had been returned for rewinding to different network voltages. This enabled the manufacturers to carry out a series of impulse tests and also to unwind the coils in order to determine exactly any defects occurring in the windings.

The smaller transformers withstood the impulse tests with full wave pulses but were damaged with chopped pulses of higher voltage. In spite of the damaged coils they withstood all the other tests to which transformers are usually subjected prior to delivery. The two large transformers went through all the impulse tests without any damage, even when impulse voltages seven times higher than the rated voltage were used.

Theoretically, the transformer can be represented by a chain of inductances and capacities.* The capacitive character is not apparent in normal operation but becomes prominent when very rapid phenomena, such as steep-fronted pulses or surges, take place. The coil stresses reach their maximum value when the front of the pulses is progressing through the windings; and another stress peak is attained thereafter as a result of the oscillations in the transformer coil, which appear in conjunction with strong magnetic fields in the secondary winding.



C_c = impulse capacitor, 9000 pF.
R_d = damping resistor.
R_e = discharging resistor.
R_t = potential divider.
C_s = load capacitor.
F_K = coupling spark gap.
F = test spark gap.
T = transformer under test.
S-O, K-O = moving-coil and cathode-ray oscillographs.

The impulse testing equipment designed for a nominal voltage of 300 kilovolts is shown in the accompanying illustration. To avoid the superposition of large amplitude oscillations along the leads to the transformer being tested, the entire testing equipment was designed as compact as possible. The a.c. excitation was achieved with normal operating frequency on the low-voltage side with full star voltage. Gas bubble formation and arc extinction were also observed, and measurements were made before, during, and after the impulse tests, to find any damage occurring in the coils. After each test or measurement the windings were investigated for insulation damage.

*P. WALDVOGEL, S. E. V. BULL, No. 3, 1942, *p.* 57.

NEWS OF THE MONTH

Announcements in this section include News relating to British Firms, and on Equipment produced by British Manufacturers.
Available literature may be secured by addressing a request to the advertising department of "The Engineers' Digest," or by writing direct to the manufacturer and mentioning "The Engineers' Digest" as a source.

- PERSONAL
- NEW EQUIPMENT
- BOOKS AND CATALOGUES
- BUSINESS CHANGES
- NOTES

PERSONAL.

Mr. F. G. Bacon, B.Sc., A.M.I.Mech.E., A.M.I.E.E., Mr. H. C. Blench, M.C., and **Mr. J. M. Graham** have been elected Directors of International Alloys Limited, Farnham Royal, Bucks. Mr. Bacon is director of production, Mr. Blench is director of sales, and Mr. Graham is a director of Almin Ltd.

Mr. R. L. Batley, A.M.I.Mech.E., has been appointed chief generation engineer; **Mr. D. F. Grant, A.M.I.E.E.,** and **Mr. H. Berry, M.Eng., A.M.I.Mech.E., A.M.I.E.E.,** senior assistant engineers; **Mr. W. J. Norrie,** mechanical maintenance engineer; **Mr. F. H. S. Brown, B.Sc., A.M.I.Mech.E., A.M.I.E.E.,** and **Mr. A. N. Duffett, B.Eng., A.M.I.E.E.,** generation engineers (construction) to the British Electricity Authority, Merseyside and North Wales Division, Clarkes Garden, Woolton, Liverpool.

Mr. G. P. Belsham, works manager of Brookhirst Switchgear Ltd., Northgate Works, Chester, has been appointed a director of the Company.

Mr. W. H. Bowman has been appointed technical director of Reynolds Light Alloys, Ltd., Reynolds Rolling Mills, Ltd., and the South Wales Aluminium Co., Ltd. **Mr. E. G. Snelus** has been appointed sales manager of Reynolds Light Alloys, Ltd., and Reynolds Rolling Mills, Ltd.

Mr. F. J. Erroll has been appointed a director of Richard Johnson and Nephew, Ltd.

Mr. A. S. Kettle, M.I.E.E., has been appointed manager of the Construction Department of The British Thomson-Houston Co., Ltd., in place of **Mr. W. J. Boddy** who retired after 51 years' service with the Company.

Mr. E. Bolton King, of the Directorate of Guided Weapons, Ministry of Supply, has been appointed Director of the Science Department of the British Council.

Mr. C. E. Knight, assistant director of the Combustion Appliance Makers' Association, 24 Tottenham Court Road, London, W.1, is now in charge of the Association's activities; Mr. P. A. H. Elliott has relinquished his position of director of the Association.

Dr. A. H. Leckie has been appointed head of the steelmaking division of the British Iron and Steel Research Association.

Group Captain P. McDiarmid, O.B.E., A.F.M., joined the Plessey Co., Ltd., Ilford, Essex.

Mr. H. S. Peiser, M.A. (Cantab), the X-Ray Crystallographer of Birkbeck College, and **Dr. W. F. Ford,** a refractories expert, of Sheffield University, have joined the Research Department of Hadfields Ltd., East Hecla Works, Sheffield. **Mr. W. H. Salmon, Assoc.Met., F.I.M.,** former President of the Sheffield branch of the Institute of British Foundrymen, has been appointed N.W. England and N. Wales Representative of the Company.

Mr. H. C. Rudd has been appointed cable engineer, and **Mr. L. R. Brock,** assistant plant and equipment engineer in the chief engineer's department of the Southern Electricity Board, Maidenhead, Berkshire. **Mr. H. Nimmo, M.I.C.E., M.I.Mech.E., M.I.E.E.,** is Chairman of the Board.

Mr. Basil Sanderson has been elected a director of Ford Motor Co., Ltd., Dagenham.

Mr. J. J. Sheehan, B.Sc., A.R.C.Sc.I., A.R.I.C., Coneygre Foundry, Tipton, Staffs., has been nominated to be junior vice-president of the Institute of British Foundrymen.

Mr. H. A. L. Trew has been appointed acting press officer of the Railway Executive (Western Region), Paddington Station, London, W.2.

Mr. E. S. Waddington, M.S.E., M.I.W., A.M.I.E., A.I.E.E., of the Industrial Department of Philips Electrical Ltd., Century House, Shaftesbury Avenue, London, W.C.2, has been elected senior vice-president of the Society of Engineers, 17 Victoria Street, London, S.W.1.

Dr. R. C. G. Williams, M.I.E.E., M.I.Mech.E., chief engineer of Philips Electrical Ltd., has been elected a Fellow of the American Institute of Electrical Engineers.

Dr. F. J. Wilkins, M.Sc., has been appointed Principal Director of Scientific Research (Defence) in the Ministry of Supply. Dr. Wilkins will be responsible for research on munitions and allied equipment.

OBITUARY.

The death occurred at his home in Seaford, Sussex, at the age of 67 of **Mr. H. A. Dormer, M.I.Mech.E.,** Chairman of The Sheffield Twist Drill and Steel Company Limited, Sheffield.

THE BRITISH INDUSTRIES FAIR.

The British Industries Fair held simultaneously at Olympia and Earl's Court in London, and at Castle Bromwich, near Birmingham, from Monday, May 3rd to Friday, May 14th, occupied a total floor space of 1,000,000 sq. ft., 90,000 sq. ft. greater than last year's B.I.F.

As Sir Graham Cunningham, K.B.E., a member of the F.B.I. and the Economic Planning Board, said the Fair provided a story which the whole world will understand—the story of skill and enterprise and courage, brain and craftsmanship. The story of a nation that three years ago was beaten to exhaustion by the bloodiest war in history—a war it won, but a war that cost more in victory than in defeat.

The Exhibition was a magnificent triumph of British enterprise and initiative, and it gave an overwhelming evidence that British engineers and manufacturers are capable of producing goods and equipment equal to and better than any other country, and of their desire to serve the needs of the world.

Buyers from more than 50 countries agreed that the Fair was well organized and they were impressed with the progressive design and the high quality of the exhibits as well as with the short delivery dates which could be offered in most cases.

BUSINESS NOTES.

Fight against Inflation. On the grounds of national policy, the Board of Imperial Chemical Industries Ltd., has decided to reduce prices over a wide range of its products. The effect of this decision will be to save the customers of I.C.I. more than £1,000,000 during the next twelve months.

The Victor Products (Wallsend) Ltd., Wallsend-on-Tyne has also decided to make an immediate and voluntary contribution to

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the national situation by reducing by 5 per cent. the prices of 99 per cent. of the Company's products.

Associated Equipment Co., Ltd., Southall, has transferred its overseas sales department to 7 Chesterfield Gardens, London, W.1.

J. H. Fenner & Co., Ltd., has opened a branch office at 86 Bridge Street, Sheffield. Mr. G. K. Varley has been appointed manager.

Mullard Wireless Service Co., Ltd., Aboyne Works, Aboyne Road, London, S.W.17, has changed the name of the Company to Mullard Electronic Products, Ltd.

Musgrave & Co., Ltd., Belfast, has opened a branch office at 89A Clayton Street, Newcastle-upon-Tyne. Mr. W. J. Adams has been appointed manager.

Specialloid Limited, North Finchley, London, N.12, have appointed the following overseas representatives:—Mr. G. W. Watts, Quinta Rosemarie, Avenida A. El Pinar, Caracas, Venezuela; Mr. Ernst Goldschmidt, P.O.B., 274 Mogadishu, Somalia; and Yangtze (London) Ltd., Imperial House, Regent Street, London, W.1, a subsidiary company of The Yangtze Development Corp., 346 Szechuen Road, Shanghai, China.

NEW EQUIPMENT.

RYALAND'S SELF-PRIMING CENTRIFUGAL PUMPS.

The illustration below shows one of the new series of Ryaland Close Coupled, Motor-Driven, Self-Priming Centrifugal Pumps, which are now being produced in various sizes for discharges up to 1,300 gallons per hour, and to operate at heads of 70 feet or less. In addition to water, they will handle light oils, emulsions, petrol, and aqueous solutions at temperatures up to 40 deg. C. All models will evacuate the air from suction pipes of 300 feet in length, and at the same time deal with a manometric suction lift of 26 feet of water or 15 feet of petrol, when running at 1,450 r.p.m. Liquid and air can be pumped separately or simultaneously. Of robust construction, these pumps occupy the minimum of floor space. The motors are above the pumps to reduce the danger of flooding to a minimum. The inlet and outlet branches are " in line," an arrangement which simplifies pipe work and installation. Gauge tappings and a low level drain plug are provided in the casings. The impellers are of hard phosphor bronze, accurately machined and balanced; the stainless steel shafts run in bearings of ample size, and particular attention has been given to the design of the double stuffing boxes, which combine accessibility for adjustment with long life and an absolute minimum of gland leakage. The motors are totally enclosed, surface cooled and available for most electrical supplies. They cannot be overloaded under normal working conditions. The pumps can be supplied to suit special requirements fitted with flame proof motors and efficient shaft seals to entirely eliminate all gland leakage.

Rigid Frame Power Presses. Investigations carried out by Messrs. Hedleys Ltd., Forward Works, West Bromwich, have shown that on a press with considerable spring in the frame the tools had to be reground after production of about 100,000 parts, in consequence of the burr on the piece part reaching a height of 0.004 in., due to the wear of the tools. After mounting the same tools in a rigid frame press, re-grinding was only necessary after production of 193,000 parts. Based on these investigations, Messrs. Hedleys Ltd. have designed high speed inclinable power presses with rigid frames, which ensure 100 per cent saving in tool costs and, in addition, considerable savings in re-grinding and setting costs and a reduction of idle times. Normal pressure of the Hedley power presses is 12½ tons, maximum pressure is 15 tons. Specification of two pressures makes it possible to choose the most suitable and efficient press for any particular purpose.

Metalastik Shackle Pin for Heavy Vehicles. A flexible shackle pin which does not require lubrication or any other attention in service for long periods has been developed by Metalastik Ltd., of Leicester.

The centre pin and outer shell, which is split into two halves, are securely bonded together by the Metalastik Rubber-to-Metal Weld process. As in all the Metalastik Patent Ultra-Duty Bushes, the rubber is pre-loaded to give greater load capacity and longer life.

The shackle pin was subjected to very stringent tests including a 500 hour test of a complete chassis under full load on the Bump-rig where every part was subjected to severe stress and strains without failure.

Electric Drive for Precious Metal Rolling Mill.

A " Brightside " mill, employed by the **Sheffield Smelting Co., Ltd.,** for rolling gold, silver (fine and alloyed), nickel and copper, has recently been equipped by **The General Electric Co., Ltd.,** with electric drive. The mill is driven by a 200 h.p. 400 volt slipring motor running at 585 r.p.m. but this speed is reduced by means of double helical gearing to about 10 r.p.m. The motor is mounted on top of the gear housing, which also incorporates the reduction gears and mill pinions.

Forced feed lubrication is provided by means of a mechanical lubricator driven by a 1 h.p. protected type G.E.C. motor, which incorporates a safety switch. Thus failure of the oil circulating system automatically stops the mill. The high-speed shaft runs in self-aligning roller bearings, while the mill pinions and slow-speed shaft are fitted with phosphor-bronze sleeve type bearings.

NEW USE FOR AIR-RAID SHELTERS.

Faced with the problem of speeding up the output from the section engaged on life-testing electrical apparatus and materials, the Development Laboratory of the **G.E.C. Witton Engineering Works** has discovered a new use for underground air-raid shelters.

As the need was urgent and the building of suitable extensions would have entailed long delay, an air raid shelter near the Laboratory has been brought into service as a test room. This is left unattended, and the equipment is left running continuously throughout 24 hours without involving any increase in personnel or causing unnecessary annoyance, thus trebling output. The performance of the equipment is checked morning and night only.

BOOK REVIEW.

" Grundgesetze der Regelung." By DR. ING. WINFRIED APPELT, published by Wolfenbütteler Verlagsanstalt, G.m.b.H., Hannover, 1947, 118 pages, 32 figs, 28 tables. Price RM10.

THIS is an excellent book on the theoretical foundations of automatic regulation. Appearing in the series " Bücher der Technik," it deals with methods of calculation only, but is to be followed by another book dealing with the application of the theory and the construction of regulating circuits. While the present volume is a " Notdruck " that is to say, it is printed on very bad paper; this should not detract from the value of the book which is useful to both students and practising engineers. The numerous figures and tables are lucidly set-out, and the excellence of the print would indeed be deserving a better quality paper.

Starting with definitions and the explanation of the aims of mathematical and graphical treatment, the book winds up with a chapter on the necessary conditions for inability of regulating circuits, and gives between the covers all that can be taught in this art. In some of its chapters it goes beyond what is considered to be general knowledge.

After having studied this book, no doubt many will be eagerly looking forward to the accompanying volume, to be entitled " Stetige Regelvorgänge."

British Standards Institution 1947 Yearbook.

This Yearbook, which has just been published by the British Standards Institution gives a synopsis of each of the 1,400 British

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Standards now current, which has been prepared by representative Committees of 50 different industries, and is obtainable from the Publication Sales Department, British Standards Institution, 24 Victoria Street, London, S.W.1.

I.D.R. Six Years Index. A comprehensive index for the first 73 issues of "Industrial Diamond Review," 1940-1946, has just been prepared and is on sale at 3s. each.

THE Index contains the following sections: Key to Page Numbers, Author's Index, Name Index, and Subject Index. Great endeavours were made to make this index as comprehensive as possible, and in particular, all names have been checked with the originals to make sure that the index gives the correct version.

ROVER GAS TURBINE ENGINE FOR CARS.

G. Geoffrey Smith, author, "Gas Turbines & Jet Propulsion for Aircraft," discloses in AUTOCAR that the Centrax gas turbine car engine shown at the B.I.F. is not the only engine of its type in existence and that the Rover Company have had an even smaller engine under actual test for over 12 months.

The Centrax engine already announced is not a show model but a prototype; indeed, it is incomplete and has not yet been run. The power turbine and the reduction gearing are not included.

The Centrax, however, is not the only example of British enterprise in the gas turbine field. The Rover Company, of Solihull, whose research section for months past has had under experimental test—and actually running on the bench—a turbine smaller than the Centrax. It is the most advanced gas turbine unit for road vehicles in the country, and probably in the world. But for the Rover policy of quietly developing new engineering designs of this character and making their announcement after full-scale trials, the public could have learned of it many months ago. Early in February, G. Geoffrey Smith inspected this unit stripped for examination after several prolonged bench tests extending from February 3, 1947. Components have run sixty hours on rig tests. The Rover Company have had exceptional experience with gas turbines for, during the war, they developed the early Whittle design to an advanced stage and, in conjunction with Joseph Lucas Ltd., incorporated "straight through" combustion, which became a feature of the highly successful aircraft gas turbines.

A most interesting design has resulted, embodying two independent turbines. One drives the air compressor, while the second turbine, running on a separate shaft, furnishes the power output. The unit, with a turbine wheel of only 5 in. diameter, develops 100 h.p. and has run at the colossal speed of 70,000 r.p.m., though its normal designed speed is 55,000 r.p.m. It scales 475 lb. complete with gear box and clutch—a saving of 200 lb. compared with the normal piston engine. It is 36 ins. long, 18 ins. wide and 20 ins. high. It has a single combustion chamber. A feature of the Rover unit is the provision of a heat exchanger, which is regarded as a necessary component to ensure a higher overall efficiency. The Rover unit is so far advanced that it could be fitted now into a car.

A new technical development of considerable importance embodied in the Rover design is that the turbine wheel, that is the disk and series of blades on the periphery, is formed in one piece, a method which should reduce production costs materially and greatly ease maintenance.

INTERNATIONAL WELDING CONGRESS.

By invitation of the Institut Belge de la Soudure, the Inaugural Meeting of a new international body, the International Institute of Welding, will be held in Brussels from 8 to 11 June, next. It is expected that 18 countries will be represented at this gathering and invitations have been sent to welding societies in all parts of the world, inviting them to send delegates to register their adhesion to the Institute. The technical meetings will also be open to all who are interested in welding.

The programme of the conference is as follows:—

One session in the afternoon of Tuesday, 8 June, will be given to technical papers by Belgian experts, and the morning and afternoon of the following day will be occupied with technical papers by non-Belgian authors. On Thursday, 10 June, there will be a number of works visits, and on Friday the Governing Council of the new Institute will hold its first meeting.

The following papers have been promised:—

On Tuesday, 8 June, "The Possibilities of Determining and Relieving Residual Stresses," by M. Walter Soote; "Scientific studies with reference to the applications of the oxy-acetylene flame," by M. Willy Bonhomme; and "The Detection of Defects in Welded Construction," by M. Georges Homes.

On Wednesday, 9 June, "The Present Position on Residual Stresses in Welded Structures," by Mr. R. Weck (Great Britain); "Oxy-Acetylene Pressure Welding," by M. C. G. Keel (Switzerland); "Evolution of Metallurgical Problems in Welding: Examples of hardening phenomena," by Professor A. Portevin and his collaborators (France); and "Mathematical Treatment yields Explanation of Brittle Fractures," by Professor F. K. Th. van Iterson (Netherlands).

The official languages of the conference are English and French.

The movement for the establishment of the International Institute began at a meeting of the Dutch Welding Society in Utrecht last summer. A Provisional Committee was appointed at a subsequent meeting in London, when 12 countries were represented. During the winter the Provisional Committee, of which the Chairman was supplied by Belgium and the Secretary by Great Britain, has formulated a draft constitution for the new body, membership of which will be open to societies and other corporate bodies of a non-commercial character, interested in welding.

Those intending to take part in the conference should communicate with the Secretary of the Provisional Committee, Mr. G. Parsloe, The Institute of Welding, 2 Buckingham Palace Gardens, London, S.W.1.

Classified Advertisements.

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CHARTER FOR TECHNICAL STAFFS in the Engineering and Metallurgical Industries. The proposals in the Charter have aroused great interest in the profession. They are being widely discussed. Any Engineer or Metallurgist may have a copy free on request.—Write to the Publishers: **The Association of Scientific Workers, 15 Half Moon Street, London, W.1.**

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